Design of a Creme-Lite Mixer (A)

The telephone rang; it was Charlie Menzie.

"George, would you go down to West's, please," he said, "they are having trouble with the creme-lite mixer that you designed."

Charlie could give no further information about the trouble. Webb, the Plant Engineer at West Ltd., had telephoned Charlie to have someone come down to see about fixing the mixer which had broken down shortly after it had been put into service. Charlie asked George to see what the trouble was and to find out what had to be done to fix it.

 Prepared for the Stanford Engineering Case Program by G. Kardos, Associate Professor of Mechanical Engineering, McMaster University, Hamilton, Ontario, Canada, with the support of the National Science Foundation. Although names of individuals and firms have been fictionalized, the case is based upon real engineering activity.

George Crassa was a professional engineer with 15 years of industrial experience. He had recently joined Roche Inc.; the Creme-lite mixer for G. West Ltd. was his first project for the company. Roche Inc. was a small steel fabricator specializing in tanks and other light gauge steel weldments. Until George joined them, Roche Inc. did not have an engineer on staff. Their expanding business and demands for original equipment by their customers had required them to increase their technical staff. A year before they had hired Harry Anson, a senior draftsman. George and Harry now comprised the entire engineering staff.

Before joining Roche Inc., George had worked in a variety of industries from a heavy machinery manufacturer to a firm making aircraft instruments. This varied background had convinced him that original design was based upon a knowledge of the fundamentals. The differences between the technologies of the different industries he had worked in were only a matter of emphasis. It was George's habit, when faced with an unfamiliar problem, to fall back on fundamentals and build his knowledge from there.

Charlie Menzie was a salesman with Roche Inc. and a personal friend of Georges. In fact, Charlie had been instrumental in recruiting George to join Roche Inc. Charlie hadn't any formal technical training but was one of those quick bright people who grasped things quickly and who was a bundle of energy.

As George drove down to West's plant, he turned the project over in his mind to see if he could anticipate what had happened.

Charlie had made contacts at G. West Ltd. They had wanted a special tank built for handling creme-lite. Roche Inc. wanted the job badly for two reasons: first it would be an introduction into a new area, and secondly, the shop was running short of work. Because of the nature of creme-lite, considerable engineering would be required in this case.

G. West Ltd. was a large commercial baker which had several plants across the country. The Montreal plant not only supplied bread, but also made a large variety of packaged cookies and cakes. They proposed to use a newly developed edible fat, creme-lite, for their icings and fillings. The proposed tank mixer was to store and process the fat.

Creme-lite is produced by Canada Packers Ltd. It had some peculiar properties, which made it especially suitable for icings; but these properties also made it somewhat difficult to handle. The fat is delivered in tank car lots at a temperature between 80° and 130°F. In this state it is a clear golden liquid. If the temperature is raised above 130°F, it burns and turns brown making it unacceptable for baking. If cooled below 80°F, it has two states. If cooled while being agitated or stirred, it forms a smooth, free-flowing, white cream-like substance, much like butter, (hence its name). If it is allowed to cool without stirring, it crystallizes and forms a hard granular fat. In the latter state it is completely unsatisfactory for inclusion into the icings or for handling.

The purpose of the mixer was to receive and store approximately 600 gallons of creme-lite at approximately 130°F. The tank was to be double jacketed so the cooling water could be used to cool the creme-lite to just under 80°F overnight. During the cooling, the creme-lite was to be continually agitated to prevent it from crystallizing. Once at the lower temperature, steam would be introduced into the jacket to mix with the water to hold it at approximately 80°F. During the day, the creme-lite was to be drawn from the bottom of the tank into the process.

Since the creme-lite was for human consumption, the tank design had to meet food handling machinery requirements. Parts in contact with the fat were to be of stainless steel. The design must be such as to prevent contaminants getting into the fat. Reentrant cavities in the fat were to be avoided to prevent accumulation of fat which would go rancid. Although the tank was to be flushed with steam between fills, reentrant corners could still cause trouble.

Charlie and Webb had made sketches of the proposed tank to determine sizes. The tank was to be approximately 5 feet, inside diameter, 5 feet high with a conical bottom. The top of the tank was to be 10 feet off the ground. The tank would be supported by 3 legs. Lids were to be provided for the top with a catwalk so that inspection and flushing with a steam hose could be carried out.

Before proceeding further with the design, Charlie and George went to see Webb at G. West Ltd. in order to establish the exact requirements of the mixer. This was most important since they were putting in a bid which was minimal, therefore Roche Inc. could not afford to have any misunderstanding as to requirements. Because Roche needed the job, the estimates had to be correct, there was no margin for error. Any additions that had to be carried out that had not been allowed for in the estimate could be the difference between a profit or loss on the job. Several hours were spent with Webb discussing detailed requirements. These were then presented to Webb in writing, which he approved (Exhibit A-1).

Of special note is item 9. Webb could not clearly establish the required mixing rate, therefore, he requested that provisions be made for changing the speed of the mixing paddles from 12 rpm to 20 rpm by some simple means such as changing belt sheaves.

Scrapers were required at the walls because it was felt that crystallization would start on the walls and if this material was not turned back into the mix it would form an insulating barrier between the cooling water and the hot fat.

It was later requested that the inner tank be designed to carry 10 psi working pressure. Webb felt that at some future date they might wish to convert the tank to a pressure vessel. For this purpose, the inner tank was to be designed to meet the ASME code for Unfired Pressure Vessels.

Webb also wanted to know the time it would take to cool the creme-lite from 130°F to 83°F. He wanted to be sure he could cool it over-night.

Before proceeding with the design, George decided to find out more about the creme-lite, since its properties were essential to the design. He and Charlie went to Canada Packers and spoke to several people about creme-lite. Canada Packers were of little help. They offered to provide all the creme-lite needed for testing, but outside of stating that its density was between .93 and .94 and that crystallization takes place at 83°F, they knew little about its physical properties. They had no viscosity figures. Nor could they explain the mechanism of crystallization or specify the mixing speeds needed to prevent crystallization.

Charlie took a sample of creme-lite to another of his customers, one in the adhesive business, to measure the viscosity of the creme-lite. George took a sample home to get the "feel" of it at various temperatures and to try to establish some mixing parameters.

Measurements established the viscosity of the creme-lite under continuous mixing at 80°F as 350 centipoise. This figure was used in all subsequent calculations. This is about the same as S.A.E. 30 oil at 68°F.

George on his part, took a 10 lb. sample of creme-lite home and on his stove did some heating and cooling tests, measuring the temperature of the fat with a laboratory thermometer. The first time he cooled the fat without mixing. The result was an extremely granular product as soon as the temperature dropped below 80°; quite comparable to lard taken out of the refrigerator. After heating it again, he allowed it to cool while stirring gently. The fat stiffened while cooling, but it was not difficult to stir. With stirring, he found it was possible to keep it smooth and creamy well below 80°F. Seconds after he stopped the mixing, the creme-lite crystallized and became hard. On re-heating, he noted that the creme-lite melted very quickly around the edges of the pot, but did not readily melt in the center. It was necessary to cut the solid core into pieces to increase the melting rate.

With this background, George started to design. First a schematic was made (Exhibit A-2). The tank proportions were established from the requirements. The outer jacket was to be 2 inches from the inner to allow for cooling water. Steam was to be introduced at the bottom.

An electric motor was selected to drive a worm gear reducer through a V belt. Wherever possible, Roche Inc. used "Croft" worm gear reducers (Exhibit A-3). The reducer was to drive the mixing shaft by means of a chain belt operating in a horizontal plane.

The bottom bearing of the mixer shaft had to be located in the creme-lite; this bearing would be of nylon. The scrapers for the wall were also to be of nylon.

The schematic layout is shown in Exhibit A-2. The selection of the gear reducer and motor would be dependent on the power consumed by the mixing. This in turn would be dependent on the mixing paddle design.

ECL 87 A

EXHIBIT A-1 REQUIREMENTS FOR MIXING TANK

roche INC.

PRESSURE TANKS & STEEL WORKS

BASIC REQUIREMENTS OF MIXING TANK FOR: G. WEST LTD.

1. Tank to be basically in accordance with Roche quotation 122-64 sketch date 8 June.

2. Cover to have two hinged lids. 304 SS

3. Inner tank to be of 304 stainless steel. Mill finish with welds ground "smooth".

4. Tank will have a working capacity of not less than 600 imp. gallons.

5. Outer shell to be of carbon sheet.

6. Platform to be provided at access ladder to permit operator to flush tank.

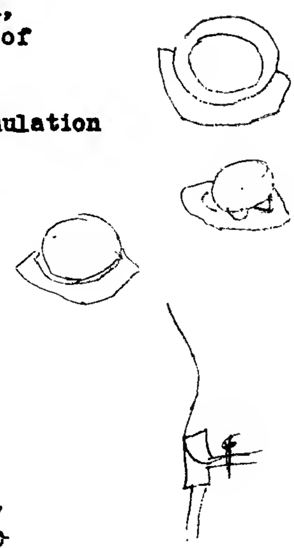
7. Food handling cleanliness requirements to be met, no corroding material to be in contact with product, contaminents must be prevented from entering processed product, no recesses or void that may trap product are to permitted.

8. Two inch radial spacing between inner and outer jacket required for circulation of water and steam.

9. Paddles to be driven at nominally 15 rpm. to be provided, paddle speed to be variable from 12 to 20 rpm by change of drive sheaves.

10. Scrapers to operate against inside wall to prevent accumulation of product on the wall.

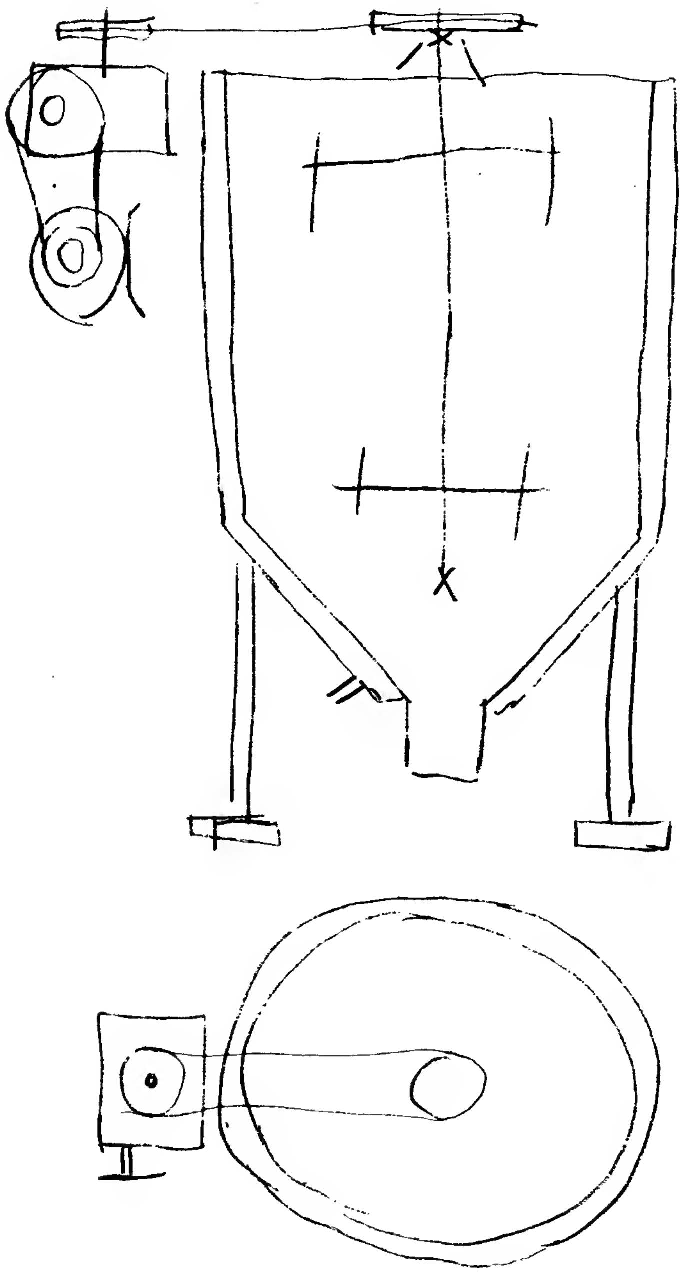
11. Connections will be of the following kind:
    1. Product inlet :- 2" MALE PIPE
    11. Product outlet :- 3" CPLG. ID 304 SS.
    111. Steam inlet. - 1" MALE PIPE
    IV. Water inlet. - 1" MALE PIPE
    V. Steam and water drain. 2"
    VI. Water drain.



12. Product will be Canada Packers "Creme - lite ".

APPROVED: A. Web July 3/64

EXHIBIT A-2 SCHEMATIC DRAWING OF MIXING TANK



Selecting a Worm Gear Reducer

When selecting a "RADIATION" Worm Gear Reducer it is necessary to determine the following factors:

(1) The maximum horsepower or torque required to drive the driven machine.

(2) The speed in R.P.M. at which the driven machine is to run.

(3) The starting-up load (this must not be under-rated).

(4) Shock loads.

(5) Any overload which may be imposed on the Worm Gear Reducer.

(6) Speed in R.P.M. of the motor or the prime mover to be used.

In determining the maximum horsepower required to drive a particular machine, the operating characteristics must be taken into account. To assist in determining the appropriate Service Factor, load and duration classifications against service factors for a variety of machinery drives are given below.

Horsepower for unit selection is obtained by multiplying the normal horsepower by the service factor for the particular type of duty for which the gear reducer is required. The resultant figure should be used to select the size of gear reducer required from the Rating Tables.

The Rating Tables tabulated on pages 4 to 8 are based on 12 hours' continuous running with temperature rise of not more than 100°F and provide the following information for each size of reducer: Nominal ratio, Exact ratio, Input shaft speed, Output shaft speed, Input H.P., Output H.P., Output torque in lbs./in.

The Thermal Rating is based on a maximum oil temperature rise of 100°F above ambient with a maximum permissible oil temperature of 200°F.

The Thermal Rating is the horsepower that can be safely transmitted by the worm gear reducer with a temperature rise of not more than 100°F above normal room temperature. Worm Gear Reducers should be selected on the basis of the mechanical rating with proper consideration of service factors, and the actual horsepower to be transmitted should be checked with the thermal rating. If the thermal rating is greater than the actual H.P. to be transmitted the Gear Reducer will operate within the allowable temperature range. If the thermal rating is less than the actual horsepower transmitted a larger size gear reducer should be used.

For all input speeds of less than 100 r.p.m. it is advantageous to use Output Torque Ratings (lbs./in.) listed, in preference to horsepowers. Since, at these low speeds, static conditions are approximated, it is not necessary to apply Service Factors.

In determining the size of reducer to accommodate shock loads, refer to the table below which gives Load Classification and duration of running time with Service Factors to be used when driving specified classes of machinery.

The permissible overhung load values, applicable at the centre of the output shaft extension, are tabulated on pages 10 and 11.

HINTS FOR SELECTING A GEAR REDUCER

STAGE 1	Decide the type of "RADIATION" Worm Gear Reducer most suitable for the proposed drive.
STAGE 2	The required ratio can be obtained by dividing the input speed R.P.M. by the R.P.M. of the driven machine (assuming the worm gear is to be coupled direct to the driven machine).
STAGE 3	Determine the input horsepower or output torque necessary for the drive, taking into account the relevant factor for the proposed duty as outlined above.
STAGE 4	Reducer size can now be determined, having decided the ratio, input speed and capacity. Refer to Output Rating Tables tabulated under input speeds ranging from 10–1800 r.p.m. and select a unit with equal or higher rating from the appropriate table.
STAGE 5	If a V-sheave, Pinion, Sprocket, etc., is to be mounted on the output shaft, check overhung load, or if vertical drive, thrust load. Permissible loads are tabulated on pages 10 and 11.
STAGE 6	Size and type of reducer now established, refer to appropriate page providing relative dimensions and shaft assemblies.

SERVICE FACTORS

CLASS OF LOAD AND DURATION OF RUNNING TIME		Electric Motor	Gas Engine Multi-Cylinder	Gas Engina Single Cylinder
UNIFORM LOAD Agitators and Mixers, liquid, semi-liquid; Conveyors, Elavators, (Buckat), Food industry, Bottling machines, etc. Pumps, gear, rotary; Screens, Air washing, travaling water.	Hours per day 3 " " " 12 " " " 24	1·0 1·0 1·25	1·0 1·25 1·5	1·25 1·5 1·75
MODERATE SHOCK LOAD Agitators and Mixers, variable density; Conveyors, not uniform load; Bucket Elevators, Food industry, slicars, mixers, grinders; Laundry Tumblers, Lineshafts, Pumps (raciprocating), Screens (rotary, stone, gravel), Taxtila machinery (batchers, calanders, mangles, soapers, spinnars, tenter framas).	Hours per day 3 " " " 12 " " " 24	1·25 1·25 1·5	1·25 1·5 1·75	1·5 1·75 2·0
HEAVY SHOCK LOAD Brick & Clay machinery (presses, briquatte machines), Conveyors (raciprocating, shakar), Metal Mills (drawbench, slittar, wire drawing), Pumps (reciprocating).	Houra per day 3 " " " 12 " " " 24	1·75 1·75 2·0	1·75 2·0 2·25	2·0 2·25 2·5

For Intermittent Service refer to Crofts for selection.

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Rating Table

THESE RATINGS APPLY TO ALL TYPES

Nominal Ratio	Exact Ratio		INPUT SPEEDS—R.P.M. 10	50	100	150	250	300	500	600	750	900	1000	1200	1500	1300
5/1	4·86/1	R.P.M. Output Shaft ...	2	10	20	30	50	60	100	120	150	180	200	240	300	360
		Input H.P.	0·68	2·3	4·2	5·7	8·5	9·6	13·5	15·3	17·8	20·4	22·0	25·0	28·3	30·5
		Output H.P.*	0·55	2·0	3·7	5·1	7·6	8·7	12·4	14·1	16·7	19·3	20·9	23·8	26·9	29·2
		Output Torque (lb. ins.) ...	17,350	12,400	11,600	10,600	9,600	9,150	7,800	7,400	7,000	6,750	6,600	6,250	5,650	5,100
7½/1	7·40/1	R.P.M. Output Shaft ...	1·33	6·66	13·3	20	33	40	67	80	100	120	133	160	200	240
		Input H.P.	0·4	2·1	3·7	5·1	7·0	8·0	11·4	12·8	14·8	16·5	17·6	19·5	22·0	24·0
		Output H.P.*	0·32	1·8	3·1	4·4	6·1	7·1	10·2	11·3	13·3	15·0	16·0	17·8	20·4	22·3
		Output Torque (lb. ins.) ...	15,000	16,500	14,850	13,800	11,650	11,100	9,560	8,900	8,380	7,890	7,570	7,000	6,440	5,850
10/1	10·33/1	R.P.M. Output Shaft ...	1	5	10	15	25	30	50	60	75	90	100	120	150	180
		Input H.P.	0·35	1·8	3·1	4·2	6·0	6·8	9·7	11·0	12·8	14·5	15·5	17·4	19·9	22·0
		Output H.P.*	0·27	1·4	2·5	3·5	5·1	5·9	8·5	9·7	11·3	12·8	13·9	15·7	18·1	20·2
		Output Torque (lb. ins.) ...	16,900	17,800	16,300	14,800	12,800	12,300	10,600	10,170	9,450	8,980	8,750	8,250	7,600	7,060
12½/1	12·33/1	R.P.M. Output Shaft ...	0·8	4	8	12	20	24	40	48	60	72	80	96	120	144
		Input H.P.	0·3	1·45	2·5	3·3	4·5	5·3	7·6	8·7	10·0	11·4	12·2	13·6	15·5	17·0
		Output H.P.*	0·22	1·1	2·0	2·7	3·8	4·6	6·6	7·6	8·8	10·1	10·9	12·2	14·0	15·3
		Output Torque (lb. ins.) ...	17,500	14,200	15,900	14,150	11,890	11,800	10,300	9,925	9,250	8,800	8,550	8,000	6,660	6,700
15/1	15·50/1	R.P.M. Output Shaft ...	0·67	3·3	7	10	17	20	33	40	50	60	67	80	100	120
		Input H.P.	0·29	1·2	2·0	2·7	4·1	4·4	6·1	6·8	7·9	9·0	9·3	10·4	12·0	13·3
		Output H.P.*	0·21	0·91	1·5	2·1	3·3	3·8	5·1	5·7	6·6	7·7	7·9	8·9	10·2	11·4
		Output Torque (lb. ins.) ...	19,900	17,390	13,850	13,490	12,300	11,700	9,700	9,000	8,390	8,030	7,425	6,970	6,425	5,970
20/1	20·50/1	R.P.M. Output Shaft ...	0·5	2·5	5	7·5	12·5	15	25	30	37·5	45	50	60	75	90
		Input H.P.	0·28	1·0	1·8	2·4	3·4	4·0	5·5	6·0	7·2	8·1	8·6	9·7	11·4	12·6
		Output H.P.*	0·2	0·74	1·4	1·9	2·7	3·2	4·5	4·9	6·0	6·8	7·2	8·1	9·6	10·6
		Output Torque (lb. ins.) ...	24,700	18,600	17,300	15,550	13,500	13,400	11,200	10,250	10,010	9,740	9,075	8,500	8,050	7,400
25/1	25·00/1	R.P.M. Output Shaft ...	0·4	2	4	6	10	12	20	24	30	36	40	48	60	72
		Input H.P.	0·27	0·8	1·4	1·8	2·9	3·3	4·5	4·9	5·8	6·6	7·0	8·2	9·7	10·1
		Output H.P.*	0·17	0·55	1·1	1·3	2·2	2·5	3·5	3·9	4·6	5·3	5·6	6·7	7·9	8·2
		Output Torque (lb. ins.) ...	26,800	17,300	17,300	13,950	13,870	13,320	11,000	10,100	9,750	9,250	8,800	8,710	8,250	7,150
30/1	31·00/1	R.P.M. Output Shaft ...	0·33	1·67	3·3	5	8·3	10	17	20	25	30	33	40	50	60
		Input H.P.	0·25	0·75	1·3	1·7	2·4	2·8	4·0	4·2	5·2	5·8	6·1	7·1	8·2	9·1
		Output H.P.*	0·14	0·47	0·89	1·2	1·8	2·1	3·0	3·2	4·0	4·5	4·8	5·5	6·4	7·2
		Output Torque (lb. ins.) ...	27,000	17,700	16,850	15,000	13,300	13,000	11,740	10,410	10,400	9,780	9,050	8,740	8,060	7,550
35/1	34·00/1	R.P.M. Output Shaft ...	0·29	1·43	2·9	4·3	7·1	8·6	14	17	21	26	29	34	43	51
		Input H.P.	0·22	0·65	1·15	1·5	2·2	2·5	3·5	4·0	4·6	5·2	5·7	6·4	7·3	8·0
		Output H.P.*	0·12	0·39	0·74	1·0	1·6	1·8	2·6	3·0	3·5	4·0	4·3	4·9	5·8	6·3
		Output Torque (lb. ins.) ...	26,300	17,200	16,500	14,650	13,800	13,200	11,650	11,100	10,490	9,590	9,410	9,100	8,450	7,740
40/1	41·00/1	R.P.M. Output Shaft ...	0·25	1·25	2·5	3·75	6·25	7·5	12·5	15	18·75	22·5	25	30	37·5	4·5
		Input H.P.	0·2	0·6	1·0	1·3	2·1	2·3	3·2	3·6	4·2	4·6	4·9	5·7	6·4	7·2
		Output H.P.*	0·1	0·34	0·59	0·81	1·4	1·5	2·2	2·5	3·0	3·3	3·5	4·1	4·7	5·3
		Output Torque (lb. ins.) ...	25,400	16,900	14,850	13,600	13,800	12,750	11,350	10,600	10,000	9,130	8,860	8,600	7,670	7,350
45/1	46·00/1	R.P.M. Output Shaft ...	0·22	1·11	2·22	3·33	5·56	6·66	11·1	13·3	16·6	20	22	27	33	40
		Input H.P.	0·19	0·5	0·9	1·2	1·9	2·2	2·8	3·2	3·8	4·3	4·6	5·1	5·7	6·6
		Output H.P.*	0·1	0·25	0·5	0·7	1·2	1·4	1·9	2·2	2·6	3·0	3·2	3·6	4·0	4·7
		Output Torque (lb. ins.) ...	26,600	14,300	14,500	13,200	13,100	13,100	10,600	10,350	9,500	9,350	9,210	8,350	7,600	7,360
50/1	49·00/1	R.P.M. Output Shaft ...	0·2	1	2	3	5	6	10	12	15	18	20	24	30	36
		Input H.P.	0·18	0·48	0·8	1·1	1·9	1·9	2·7	3·0	3·4	3·8	4·1	4·7	5·4	5·9
		Output H.P.*	0·09	0·24	0·42	0·6	1·1	1·2	1·7	2·0	2·3	2·8	2·8	3·2	3·7	4·1
		Output Torque (lb. ins.) ...	26,700	15,100	13,090	12,500	14,100	12,400	10,500	10,400	9,600	9,000	8,650	8,500	7,800	7,250
60/1	60·00/1	R.P.M. Output Shaft ...	0·16	0·83	1·6	2·5	4·2	5	8·3	10	12·5	15	16·6	20	25	30
		Input H.P.	0·13	0·375	0·68	0·9	1·3	1·6	2·2	2·5	2·9	3·3	3·4	3·8	4·4	4·9
		Output H.P.*	0·06	0·18	0·35	0·49	0·74	0·94	1·4	1·6	1·9	2·2	2·3	2·6	3·0	3·4
		Output Torque (lb. ins.) ...	23,600	14,000	13,700	12,300	11,100	11,800	10,580	10,200	9,700	9,430	8,630	8,120	7,660	7,100
70/1	71·00/1	R.P.M. Output Shaft ...	0·14	0·71	1·4	2·14	3·5	4·3	7·1	8·6	10·7	12·8	14·3	17·2	21	26
		Input H.P.	0·1	0·32	0·6	0·8	1·2	1·4	2·0	2·3	2·5	2·8	2·9	3·3	3·9	4·3
		Output H.P.*	0·05	0·18	0·3	0·43	0·67	0·81	1·3	1·5	1·7	1·9	1·9	2·2	2·7	2·9
		Output Torque (lb. ins.) ...	20,200	13,900	13,500	12,500	12,050	11,850	11,200	10,780	9,700	9,100	8,550	8,200	7,960	7,090

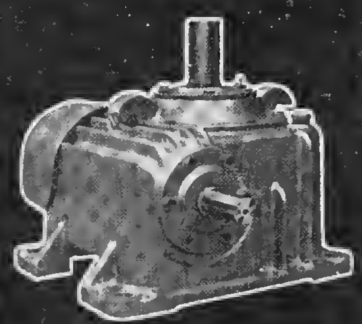
* Mechanical Ratings are equal to or less than the Thermal Ratings.
R.P.M. of output shaft are based on Nominal ratios. To obtain exact R.P.M. of output shaft, divide R.P.M. of input shaft by exact ratio.

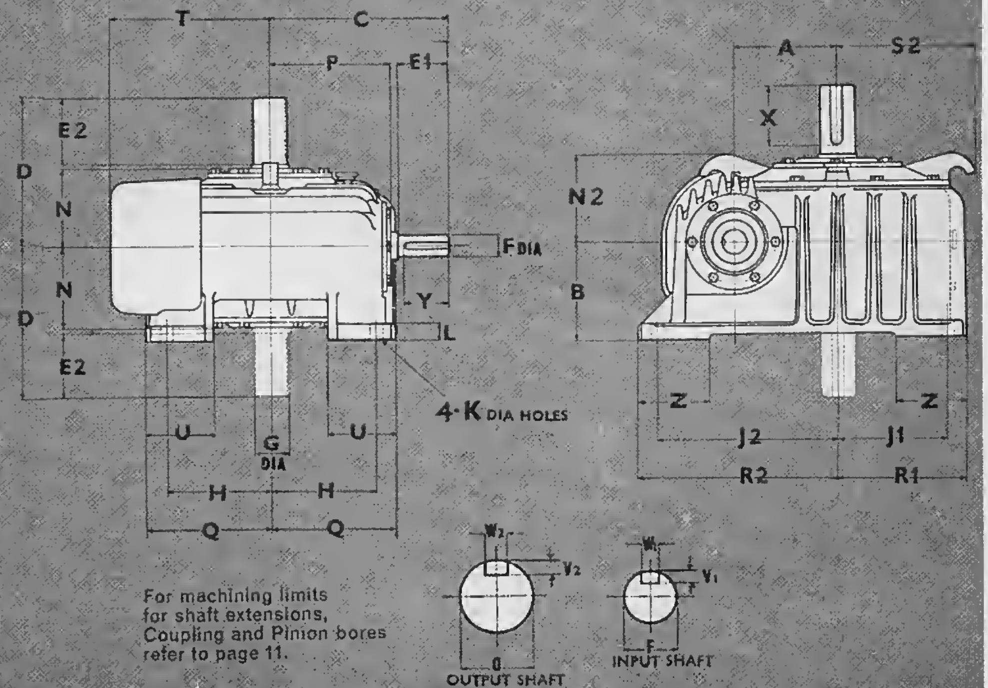
Powers listed are based on 12 hours' continuous running with temperature rise of not more than 100°F.

Units will carry 100% momentary overload on starting.

DIMENSIONS
VERTICAL TYPE

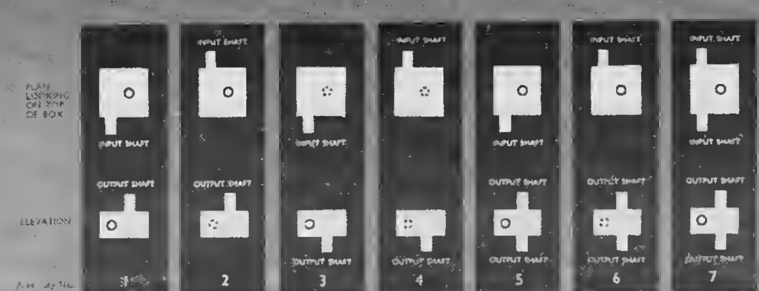
'RV' TYPE
FAN COOLED





DIMENSIONS IN INCHES

Size	A	B	C	D	E_1	E_2	F	G	H	J_1	J_2	K	L	N	N_2	P	Q	R_1	R_2	S_2	T	U	W_1	V_1	W_2	V_2	X	Y	Z	Net Weight lbs.
40	4	5	9	$8\frac{1}{2}$	$2\frac{5}{8}$	$3\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{3}{4}$	$4\frac{7}{8}$	$4\frac{7}{8}$	$7\frac{7}{8}$	$\frac{13}{16}$	$\frac{3}{4}$	5	$4\frac{3}{4}$	$6\frac{1}{4}$	6	6	9	$5\frac{7}{8}$	$8\frac{3}{8}$	$3\frac{1}{2}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{3}{8}$	$\frac{3}{16}$	$3\frac{3}{8}$	$2\frac{1}{2}$	$3\frac{1}{4}$	150
50	5	$5\frac{3}{4}$	$10\frac{1}{4}$	$9\frac{3}{4}$	$2\frac{7}{8}$	4	$1\frac{1}{2}$	2	$5\frac{7}{8}$	$5\frac{7}{8}$	$9\frac{3}{8}$	$\frac{13}{16}$	$\frac{3}{4}$	$5\frac{5}{8}$	$5\frac{3}{4}$	$7\frac{1}{4}$	7	$6\frac{7}{8}$	$10\frac{3}{8}$	$6\frac{3}{4}$	$9\frac{5}{8}$	$4\frac{1}{8}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{1}{4}$	$3\frac{7}{8}$	$2\frac{3}{4}$	$3\frac{3}{8}$	210
60	6	$6\frac{1}{2}$	11	$10\frac{3}{4}$	3	$4\frac{1}{2}$	$1\frac{1}{2}$	$2\frac{1}{4}$	$6\frac{3}{8}$	$6\frac{3}{8}$	11	$\frac{15}{16}$	1	$6\frac{1}{4}$	$6\frac{1}{8}$	$7\frac{7}{8}$	$7\frac{3}{4}$	$7\frac{5}{8}$	$12\frac{3}{8}$	$7\frac{7}{8}$	$10\frac{3}{8}$	$4\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{1}{4}$	$4\frac{3}{8}$	$2\frac{7}{8}$	$3\frac{5}{8}$	305
70	7	7	$12\frac{1}{2}$	$11\frac{3}{4}$	$3\frac{3}{8}$	5	$1\frac{3}{4}$	$2\frac{1}{2}$	$7\frac{1}{4}$	$7\frac{1}{4}$	$12\frac{3}{4}$	$\frac{15}{16}$	$1\frac{1}{8}$	$6\frac{3}{8}$	$6\frac{1}{4}$	9	$8\frac{7}{8}$	$8\frac{3}{4}$	$14\frac{1}{4}$	9	$11\frac{1}{2}$	$4\frac{7}{8}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{5}{8}$	$\frac{5}{16}$	$4\frac{7}{8}$	$3\frac{1}{4}$	$4\frac{1}{4}$	435
80	8	$7\frac{1}{2}$	$13\frac{1}{2}$	$12\frac{1}{4}$	$3\frac{1}{2}$	$5\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{3}{4}$	$8\frac{1}{2}$	$8\frac{1}{2}$	14	$1\frac{1}{16}$	$1\frac{1}{4}$	$6\frac{1}{2}$	$6\frac{3}{4}$	$9\frac{3}{4}$	$10\frac{1}{8}$	10	$15\frac{1}{2}$	$10\frac{3}{4}$	$12\frac{3}{4}$	$5\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{5}{8}$	$\frac{5}{16}$	$5\frac{3}{8}$	$3\frac{3}{8}$	$5\frac{1}{2}$	520



REDUCER ASSEMBLY ARRANGEMENTS

Output Shaft Extension—Standard Assembly No. 1—Output Shaft projects in an upward direction and is placed on Right-hand side when looking at the Input Shaft End.

NOTE.—Output Shaft can be arranged upwards or downwards, or both. If other than Standard Arrangement is required, please state Arrangement (No. 2, 3, 4, 5, 6 or 7) on enquiry and order.

Engineering Case Library

ECL 87B

Design of a Creme-Lite Mixer (B)

George first checked to see that 600 gallons of creme-lite would fit into the tank. (His calculations for the tank design are in Exhibit B-1.) The tank dimensions, 5 ft. diameter, ft. high, with 2 ft. high cone at the bottom was found to be satisfactory. In fact, this left a comfortable freeboard of 9 inches at the top.

Since the tank had to be capable of carrying 10 psi pressure, wall thickness was determined from thin cylinder theory. Nominal design indicated 1/8 inch sheet stainless would be satisfactory. Re-checking against ASME code increased the wall thickness to 3/16 inches. The ribs in the conical portion were designed to carry the total weight of the fat in the tank.

These calculations established the structural requirements of the tank. The power requirements would depend on the paddle design and the resultant drag on the paddles. As a starting point, a paddle arrangement was selected arbitrarily (Exhibit B-2). The arrangement was considered a maximum required. The paddles were arranged to insure that the entire volume was swept. Sketches were also made of the wall scraper arrangement. This paddle arrangement was used in the subsequent analysis. From elementary fluid dynamics, it was known that the drag D was

$$D = \frac{C_D A \rho V^2}{2}$$

where

C_D = drag coefficient
A = projected area
ρ = fluid density
V = fluid velocity

 Prepared for the Stanford Engineering Case Program by G. Kardos, Associate Professor of Mechanical Engineering, McMaster University, Hamilton, Ontario, Canada, with the support of the National Science Foundation. Although names of individuals and firms have been fictionalized, the case is based upon real engineering activity.

The drag coefficient is dependent on Reynolds number R. Assuming the velocity of the paddle would be the velocity at a mean diameter, 4 ft., the Reynolds number was found to be 1,000. The paddle sections were to be curved plates. The drag coefficient for an infinite flat plate or an infinite circular cylinder at R = 1,000 would be 2 and 1.5 respectively.

Therefore, the total drag on a 1 ft. by 6 ft. paddle was determined. The resultant power required at 20 rpm was found to be 1.5 Hp.

If the same paddle is at 2 ft. mean radius, the power required is reduced to .18 Hp.

This difference immediately emphasized the fact that the power requirements and the drag force were extremely dependent on the paddle location. It also indicated that using average velocities for the drag was not too accurate.

A general equation expressing the power requirements for the paddles was derived

$$Hp = \frac{C_D \rho A}{66,000} \left(\frac{\pi}{60}\right)^3 \times d^3 \times n^3$$

For the creme-lite, this equation reduced to

$$Hp = 0.255 \times A \times \left(\frac{d^3}{4}\right) \left(\frac{n^3}{20}\right)$$

This equation also showed that power requirements dropped off very rapidly as the speed was reduced from the maximum of 20 rpm.

This formula was used to esimate the power needed and to select the drive units. Since the mixer could be run at 20 rpm, the selections were made for this rating. The scrapers were assumed to have the same drag as the paddles, but acting at 5 feet.

The preliminary calculations indicated a requirement of 1.633 Hp. Allowing for extra drag an paddle arms and any center paddles the total requirement should be less than 2 Hp. It was decided to motorize at 3 Hp. since losses in the gear box and chain drive were expected.

Although the details of the paddles had not been established, the above power calculations were sufficient for selecting the drive motor and the gear box. It was essential to specify these early in the design because they were long delivery items and had to be ordered immediately. From the Crofts catalog the "RV" type fan-cooled reducer size 50 with a nominal ratio of 60/1 was chosen. This has a 2.6 Hp. output at 20 rpm with a 3.8 Hp. input at 1200 rpm.

George called the motor supplier to see if a 4 Hp. 1200 rpm motor was available. They could supply one if necessary, but it would be special; the standard NEMA ratings were 3 Hp. and 5 Hp. Since the standard motor was cheaper, it was decided to use a 5 Hp. 1200 rpm 60 cycle three phase motor.

The drive between the motor and the gear box was selected as a V-belt with 1 to 1 pulleys at 20 rpm. The unit would be supplied with 3 to 4 pulleys initially, for a 15 rpm mixing speed.

Having ordered the drive components, George turned his attention to the detail paddle design. The problem was now to design the most efficient paddle arrangement within the power limitations of the drive component. The previous derivation established the nature of the drag forces and indicated that the arrangement of paddles in Exhibit B-2, although apparently symmetrical on paper, actually was not dynamically symmetric.

George recalled, "I tried several paddle arrangements on a sketch pad. I had made a brief literature search to see if I could get any guidance but I could find nothing. In these various trials, the controlling design philosophy began to

form in my mind. The best design would be one with balanced loading. The dynamic forces on the paddles should be balanced to virtually eliminate the bearing loads." To this end George decided to make the design such that:

1. the paddle should not produce radial thrust,
2. the force on the right and left hand paddles should be equal,
3. the total force on the top set of paddles and bottom set of paddles should be equal.

The first requirement was met by having the paddles slightly bowed, but symmetrical about the direction of motion.

The second requirement could be satisfied by symmetrical construction, but this would have resulted in either large drag with too many paddles or channeling at the paddle radius. Instead it was decided to use two different sizes of paddles at different radii sweeping the entire volume.

The third requirement could be met by having identical paddle arrangements top and bottom except displaced 180°.

It had been previously established that the drag force was dependent on the radius. To establish the size of the paddles to meet the second requirement, a more careful analysis of drag forces was required. It was assumed that the drag on an element of the paddle was dependent only on the velocity. An analysis was carried to determine the relative width of the paddles to have equal drag, (Ex. B-3).

It was found that the drag force was equal to:

$$F = C\,(\ell o^{3} - \ell i^{3})$$

F = drag force
C = lumped constants
ℓo = outside radius of paddle
ℓi = inside radius of paddle

After some juggling of dimensions, it was decided to use three sets of paddles.

A small paddle, 4 inches wide, 1-1/2 feet high with its outer edge 2 feet from the center; a larger paddle, 6 inches wide by 2 feet high with outer edge at 1 ft. 6-3/8 inches from the center. In addition, perforated full length panels 6 inches wide would be attached on both sides of the center shaft at right angles to the main paddle assembly. (This paddle arrangement is shown in Exhibit B-4).

The power requirements for these paddles were checked and found satisfactory. The size of the torque tube was then determined. (Exhibit B-5)

The remaining details were cleaned up by selecting a Renold single strand chain ASA size 80 with appropriate sprockets. A flange type cartridge bearing was selected for the top bearing and a nylon machined sleeve bearing was used as bottom bearing.

The design was essentially completed. George turned the design over to Harry Anson. Harry undertook the detail design. George explained the requirements and gave him two sketches; the general layout, showing the details of the bearing dirt seal and details of the shaft torque tube attachment; and the paddle arrangement drawing with the method of attaching the scrapers (Exhibit B-4).

With these and the dimension drawings of purchased hardware (Exhibit B-6), Harry proceeded with a layout. When the layout was completed, George checked the design; little change was required. From the layout, Harry proceeded with detailing.

Before the detailing was completed, Webb from West's Ltd. asked that two thermometer wells be added -- one in the water and one in the creme-lite. The required wells were approximately 18 inches long and 1-1/8 inch iron pipe size, lawler separate thermo wells to take type "S" bulb.

EXHIBIT B-1 TANK DESIGN CALCULATIONS

VOLUME : 600 IMP GALLONS : 5400 # of CREME-LITE
= 600 x 0.1605 : 96.2 cu ft.

INSIDE DIA : 5' x ~~5' Cylinder~~ +
Cylinder HT. : 5'
Cone HT : 2'

VOLUME OF TANK: $5^2 \times \frac{\pi}{4} \left(5 + \frac{2}{3}\right)$

$= \frac{\pi}{4} \times 25 \times 5.666 = 111$ cu ft

FREE BOARD : $(111 - 96.2) \times \frac{4}{\pi \times 25} = .756$ ft ≈ 9"

STATIC PRESSURE : $(60 + 24) \times 62.4 \times 5.787 \times 10^{-4}$
$= 84 \times 62.4 \times 5.787 \times 10^{-4}$
= 3.2 psi.

ADDITIONAL ~~WORKING~~ OPERATING PRESSURE = 10 psi
MAXIMUM ALLOWABLE WORKING PRESS. = 15 psi
DESIGN PRESSURE = 15 x 2 = 30 psi.

$S = \frac{P \times d}{2t}$

$t = \frac{P \times d}{2S}$

MAT'L = 304 STAINLESS STL.
YIELD STRENGTH = 30,000 psi,
ASME DESIGN STRESS = 11.2 KSI.
ASSUME JOINT EFF = 100%

$t = \frac{30 \times 60}{2 \times 11{,}200} = .0805"$

IF US 1/8 SHEET JOINT EFFY = $\frac{.0805}{.125} \approx 60\%$

$S_0 = \frac{30 \times 60}{.25} = 7200$ psi.

EXHIBIT B-1 (Cont.)

ASME CODE FOR CLASS 3 VESSELS
SINGLE-WELDED BUTT JOINTS LESS THAN $\frac{1}{4}$ in THICK
USES DESIGN STRESS 5600 psi.

$$t = \frac{30 \times 60}{2 \times 5600} \quad \frac{900}{5600} = .16 \text{ in.} \quad \approx \frac{5}{32}.$$

$\frac{3}{16} = .1875$ in.

RIBS ON TANK CONE

NO OF RIBS = 9

LOAD PER RIB = $\frac{5400}{9} = 600^{\#}$

IF ONLY 3 LEGS LOAD/LEG = $\frac{5400}{3} = 1800$ say $2000^{\#}$

$$S = \frac{bd^2}{6} = \frac{.5 \times (2)^2}{6} = \frac{.5 \times 4}{6} = \frac{2}{6} = \frac{1}{3} = .333$$

$$\sigma = \frac{Mc}{I} = \frac{M}{S} = \frac{600 \times \cancel{30}\ 30}{.333} = \frac{18000}{.333}$$ CONCENTRATED LOAD

$$\sigma = \frac{M}{S}$$

$$M = \frac{wl^2}{2} = \frac{600 \times 30}{2} = \cancel{9000}\ 9000 \text{ in lbs}$$

$$\sigma = \frac{9000}{.333} = 27000 \text{ psi}$$ - Uniformly Loaded

Uniformly increasing load

$$M = \frac{Wl}{3} = \frac{600 \times 30}{3} = 6000 \text{ in lbs}$$

$\sigma = 18000$ psi

$\sigma < 18,000$ psi.

EXHIBIT B-1 (Cont.)

Drag: ~~A~~ $D = C_D \frac{A \rho V^2}{2}$

A: Projected Area
ρ: fluid density
V: velocity
C_D: drag coefficient

μ: viscosity Creme-lite μ = 350 centipoise approx SAE 30 Oil @ 68°F
$\mu = \frac{350 \times .000672}{32.2}$ slugs/sec ft $= .0073$

R = Reynolds Number $= \frac{\rho V L}{\mu}$

$\rho = \frac{SLUGS}{FT^3}$ $V = \frac{FT}{SEC}$ $L = FT$

$\mu = \frac{SLUGS}{FT^3} \times \frac{FT}{SEC} \times FT = \frac{SLUGS}{FT\ SEC.}$

$\rho = \frac{62.4}{32.2} \times .9 = 1.75$ SLUGS/FT^3

Lo L = 1 ft
$V = \frac{\pi \times 4 \times 20}{60} = \frac{\pi}{3} \times 4 = 4.2$ ft/sec.

$R = \frac{1.75 \times 4.2 \times 1}{.0073} = 1000 = 10^3$

$C_D = 2$ infinite flat plate = 4" any width
$C_D = 1.5$ infinite circular cylinder 1" dia.

EXHIBIT B-1 (Cont.)

ASSUME Paddle 1' x 6' Area A = 6 ft^2

$$D = \frac{2 \times 6 \times 1.75 \times (4.2)^2}{2}$$

$D = 1.75 \times 6 \times 17.5 = 183$ lbs. - SAY 200#

TORQUE = $200 \times 2 = 400$ ft#

@ 20 rpm

$$HP = \frac{200 \times \pi \times 4 \times 20}{33000} = \frac{16 \times \pi}{33} = 1.525 \text{ Hp.}$$

ASSUME 80% EFF. of DRIVE = 1.91 hp.

IF ALL OF PADDLE IS AT 1 ft

$$V = \frac{\pi \times 2 \times 20}{60} = \frac{\pi}{3} \times 2 = 2.1 \text{ ft/sec}$$

$$D = \frac{2 \times 6 \times 1.75 \times (2.1)^2}{2} = 45.7\#$$

TORQUE = $45.7 \times 1 = 45.7$ ft#

$$HP = \frac{45.7 \times \pi \times 2 \times 20}{33,000} = .18 \text{ HP.}$$

EXHIBIT B-1 (Cont.)

$$D = \frac{C_D A \rho V^2}{2}$$

$$HP = \frac{D \times V}{33,000} = \frac{C_D A \rho V^3}{2 \times 33,000}$$

$$V = \frac{\pi \times d \times rpm}{60}$$

$$Hp = \frac{C_D \rho}{2 \times 33,000} A \left(\frac{\pi \times d \times rpm}{60} \right)^3$$

$$HP = \frac{C_D \rho A}{2 \times 33000} \times \left(\frac{\pi}{60} \right)^3 \times d^3 \times rpm^3$$

with $C_D = 2$ $\rho = 1.75'$ slugs/ft^3

$$\boxed{HP = .255 \times A \times \left(\frac{d}{4} \right)^3 \times \left(\frac{rpm}{20} \right)^3}$$

EXHIBIT B-1 (Cont.)

$D_o = 4'$

$D_n = \frac{4}{\cancel{}} = \cancel{} \quad \frac{4}{1.59} = 2.5 \qquad A = 2 \times 2 \times 1.75 = 7$

$HP = .255 \times 7 \times \left(\frac{4}{4} \times \frac{1}{4}\right)\left(\frac{2.5}{4}\right)^3 \times \qquad$ @ 20 rpm

$= .255 \times 7 \times (.625)^3$

$= .255 \times 7 \times .242$

$HP_{e\,20\,rpm} = .433$ H.P. (Paddles

Scrapers $D = 5' \quad A = 2\frac{1}{4} \times 5 =$

$HP = .255 \times 2\frac{5}{4} \times \left(\frac{5}{4}\right)^3$

$HP_{20} = .255 \times 2.5 \times 1.88$

$HP_{20} = 1.2$

$\therefore$ TOTAL IN MAIN TANK: $1.2 + .433 = 1.633$ H.P.
SAY 2 H.P. FOR TOTAL TANK.

MOTORIZE AT 3 H.P.

EXHIBIT B-2 PRELIMINARY SKETCH OF PADDLE ARRANGEMENT

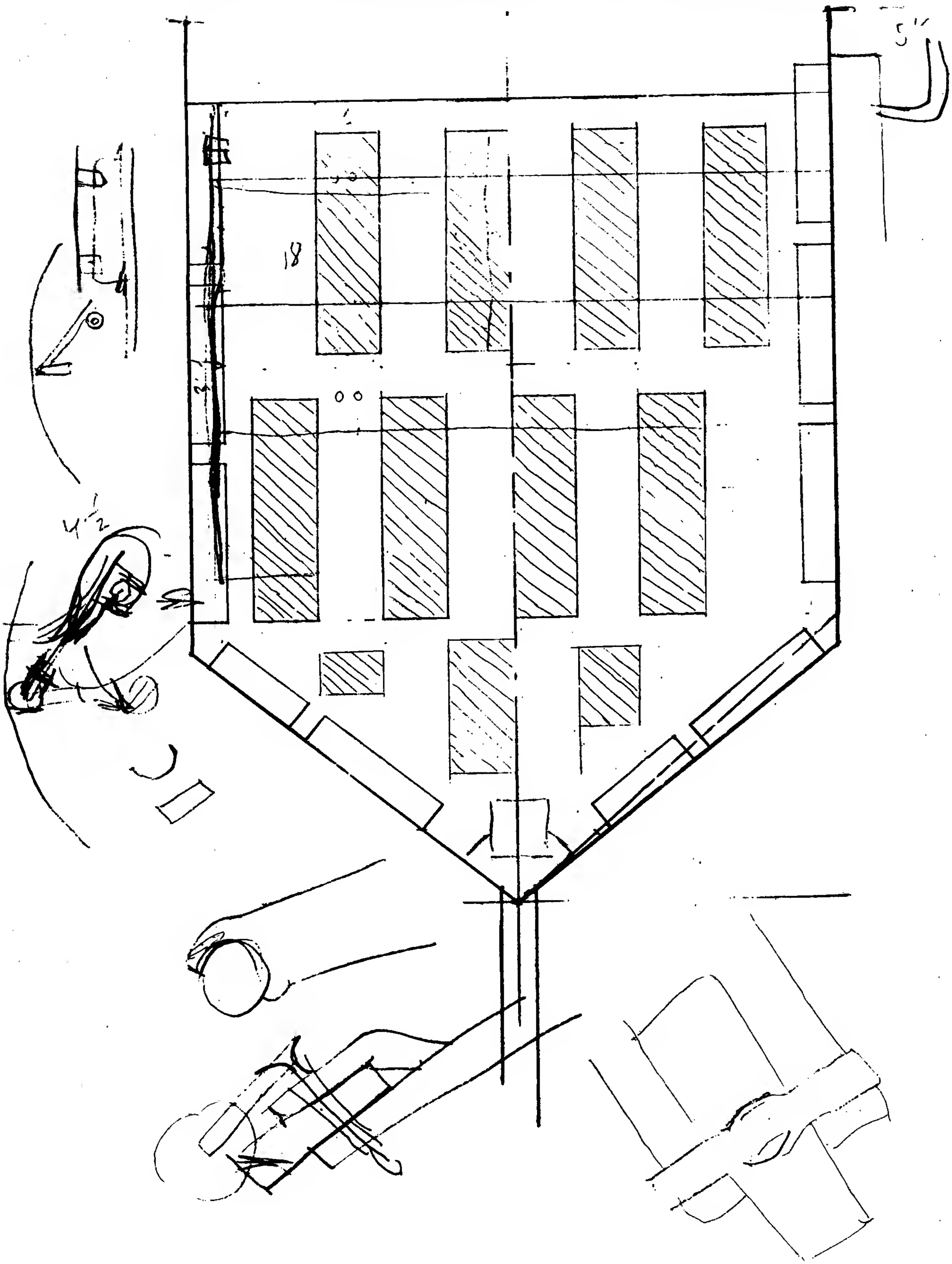
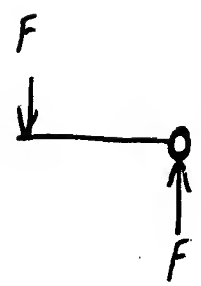


EXHIBIT B-3 ANALYSIS OF DRAG FORCES



TO MAKE DRAG ON INNER AND OUTER MIXING ~~BAR~~ PRODUCT TO SAME.

$$F = C_d \rho A V^2$$

$$V = \pi d \times rpm$$

$$F = C_D \rho A (\pi d\, rpm)^2$$

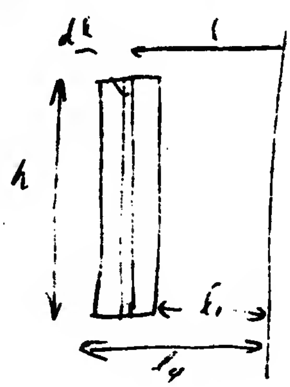
K = CONSTANTS FOR THIS APPLICATION AND PURPOSE

$$F = K' A d^2$$

$$dF = K h l^2 \, dl$$

$$F = Kh \int_{l_1}^{l_2} l^2 \, dl$$

$$= \frac{Kh}{3} \left[l^3 \right]_{l_1}^{l}$$



$$= \frac{Kh}{3} \left[l_4^3 - l_1^3 \right] = C \left(l_2^3 - l_1^3 \right)$$

For total plate $l_4 = 4$, $l_1 = 1$; $l_4^3 = 64$, $l_1^3 = 1$

$$(l_4^3 - l_1^3) = 63$$

inner panels start at 1 ft.

Four panels each with $\frac{63}{4} = F/C$

$$\therefore l_2^3 - l_1^3 = \frac{63}{4} = l_2^3 - 1$$

$$l_2 = \frac{63 + 4}{4}$$

EXHIBIT B-3 (Cont.)

$$\ell_2^3 = \frac{67}{4} = 16.75$$

$$\ell_2 = \sqrt[3]{16.75} = 2.56$$

$$(\ell_3^3 - \ell_2^3) = \frac{67}{4}$$

$$\ell_3^3 - 16.75 = \cancel{\frac{67}{4}}\ \frac{63}{4} +$$

$$\ell_3^3 = \frac{63}{4} + 16.75 = \frac{63}{2}$$

$$\ell_1 = 1.5'$$

$$\ell_3^3 - \ell_1^3 = 64 - 3.38$$

$$\ell_3 = 4 \qquad \ell_1 = 2$$

$$F = C\,(64 - 8) = C\,56$$

$$\frac{F}{2} = C\,\frac{56}{2}$$

$$\frac{F}{2} = C\,\frac{56}{2} = C\,(\ell_2^3 - \ell_1^3)$$

$$\frac{56}{2} + 8 = \ell_2^3$$

$$\sqrt[3]{36^3}$$

$$\ell^2 = \sqrt{36} = 3$$

$$(\ell_2^3 - \ell_1^3) = \frac{C\,\frac{56}{2}}{C} =$$

$$(\ell_2^3 - \ell_1^3) = \frac{56}{2} \times \cancel{}\ \cancel{\frac{16}{24}}\ \frac{18}{24}$$

$$= 21$$

EXHIBIT B-3 (Cont.)

$l_2^3 = 21 + 8$

$l_2 = \sqrt[3]{29}$

$$\begin{array}{r} 12 \\ .06 \\ \hline .72 \end{array}$$

3.06 $3\frac{3}{4}$

$3' 0\frac{3}{4}''$

$1\frac{3}{8}$

0 $1\frac{3}{8}$

1 $6\frac{3}{8}$

EXHIBIT B-4 SKETCH OF PADDLE ARRANGEMENT

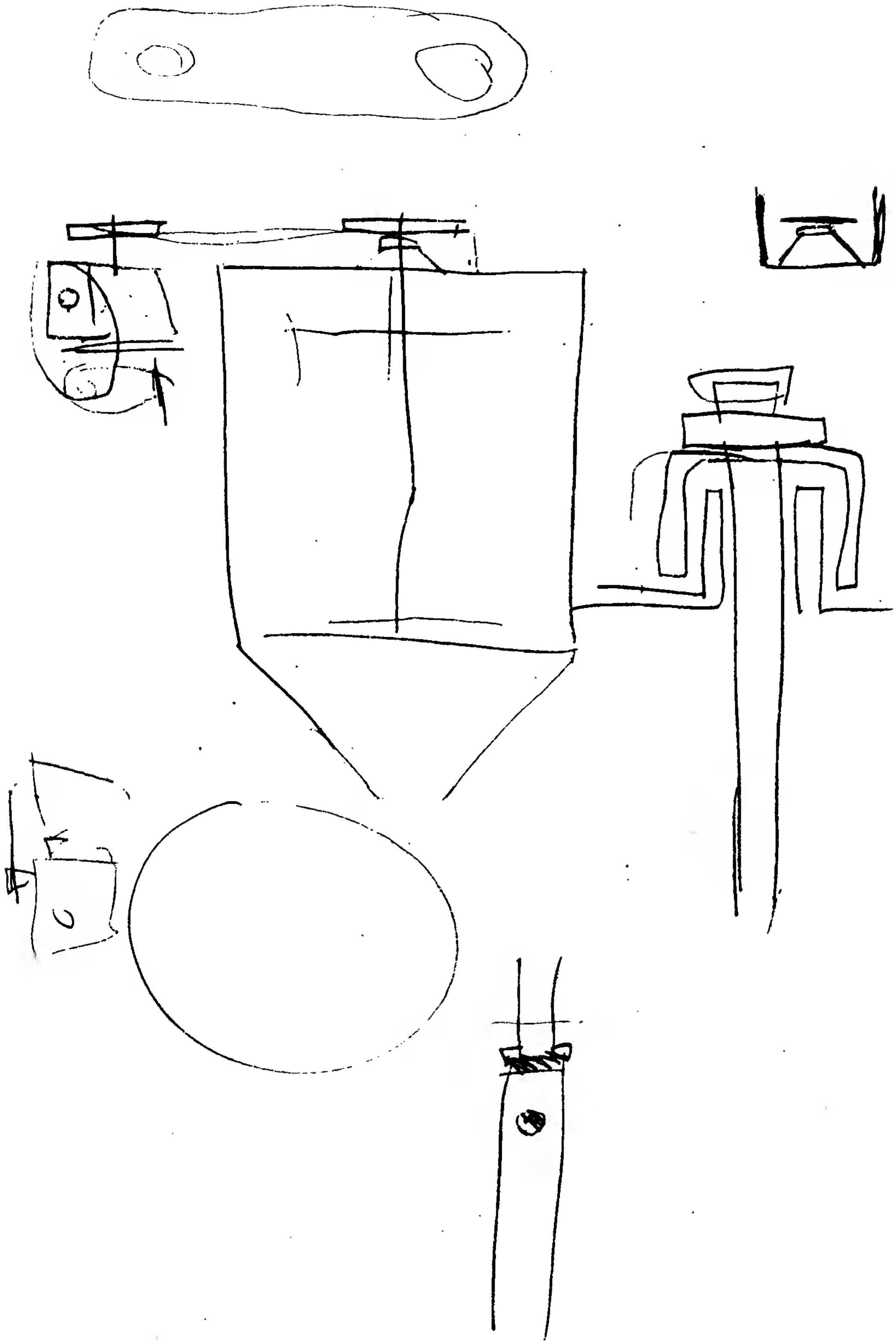


EXHIBIT B -4 (Cont.)

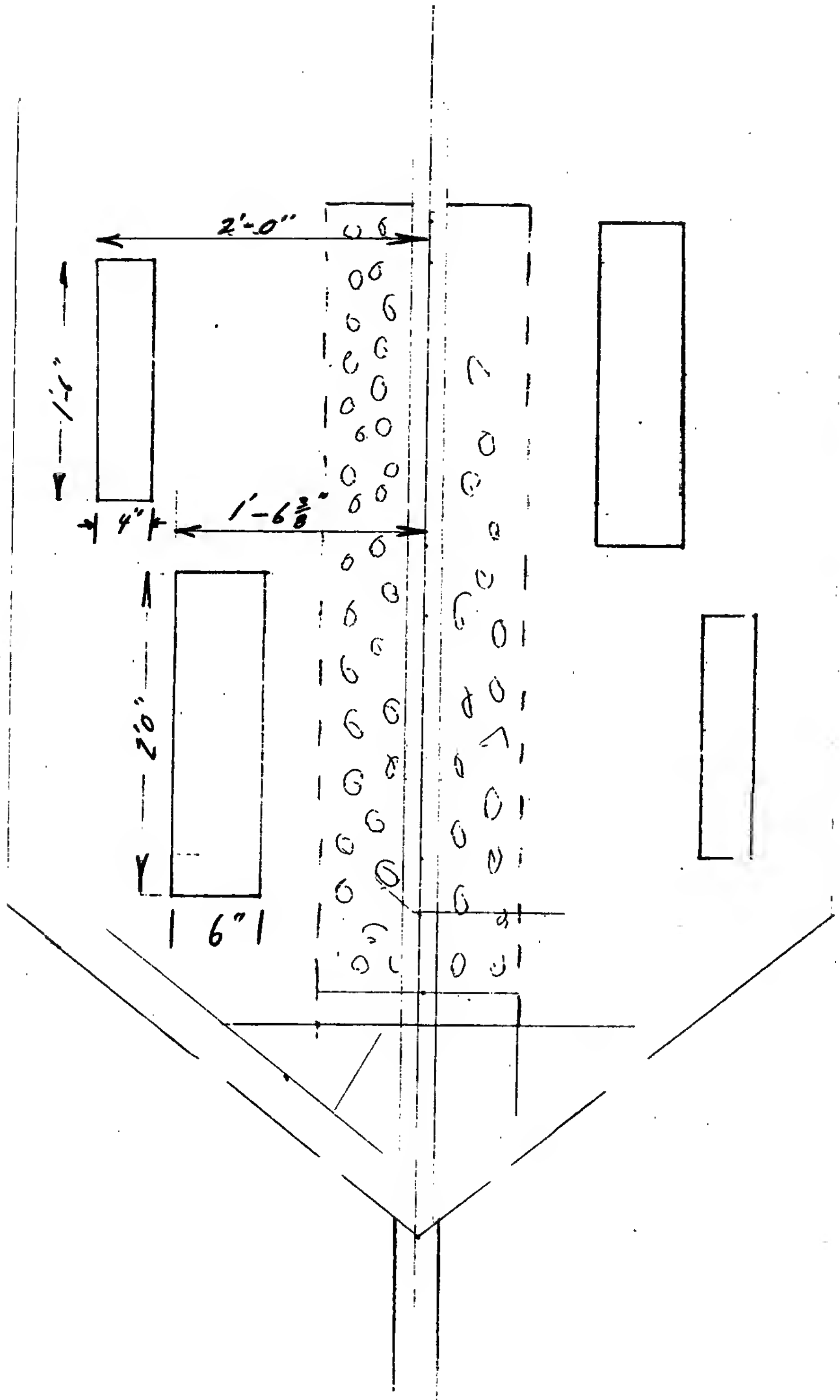


EXHIBIT B-5 DETERMINATION OF TORQUE TUBE SIZE

$$H.P. = .255 \times A \times \left(\frac{d}{4}\right)^3 \times \left(\frac{rpm}{20}\right)^3$$

SCRAPERS $A = \frac{1}{4} \times 5' \times 2 = 2.5\ ft^2$

$d = 5'$ $\frac{d}{4} = 1.25$

$\left(\frac{d}{4}\right)^3 = 1.25^3 = 1.98$

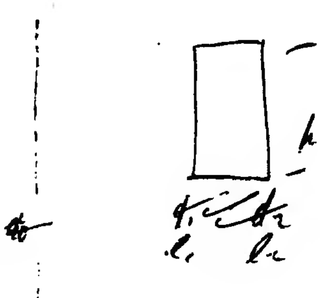
$HP = 1.25 \times 1.98 \times .255 = .63\ H.P.$

OUTTER PADDLE

$$HP = \frac{.255}{4^3} \times A\left(d^3\right)$$

$$\frac{.255}{4^3} A d^3 = \frac{.255}{64} \times A d^3$$

$$= .00399\ A\ d^3$$ then (ft)



$HP = .00399 \times h\ dl\ l^3$

$d(HP) = .00399 \times h\ l^3 dl$

$$HP = .00399 \times h \int_{l_1}^{l_2} l^3 dl = .00399\ h\ \frac{1}{4}\left[l^4\right]_{l_1}^{l_2}$$

EXHIBIT B-5 (Cont.)

$$HP = \frac{.00399}{4} h \left(l_2^4 - l_1^4\right)$$

let $h = 1.7$ ft

$$HP = \frac{.00399 \times 1.7}{4} \left(l_2^4 - l_1^4\right)$$

$l_4 = 4$ $\quad l_4^4 = 256$

$$HP = \frac{.00399 \times 1.7}{4} \times 256 = .425 \text{ hp.}$$

Total Paddle + Scrapper

$= .825 + .67$

If each segments uses $.1$ hp. or $\frac{.425}{4}$

inner paddle $\sqrt[4]{\frac{256}{4}} = 64$

$d = 283$ $\quad$ (2.78

EXHIBIT B-5 (Cont.)

3 H.P. @ 20 rpm. Chain Drive

≈ 10" Dia. Chain velocity =

$\pi D \times 20 = \pi \times \frac{10}{12} \times 20 = 52.3$ ft/min.

$$3 = \frac{L \times 52.3}{33,000}$$

$$L = \frac{3 \times 33,000}{52.3} = 1,900\#$$

Minimum Chain 3/4" Pitch single 4/ , 2000 lbs

Chain No 80
Pitch 1" Roll dia .625
Breaking load 14,500 Wt/foot 1.68

Load Capacity at 25 rpm 2.639 with 36 Teeth

EXHIBIT B-5 (Cont.)

Torque Tube.

MAT'L 304 STAINLESS

O.D. : $2\frac{3}{8}$"

I.D. : 1.93"

Torque 8,500 in #

$$\tau = \frac{T r}{J} =$$

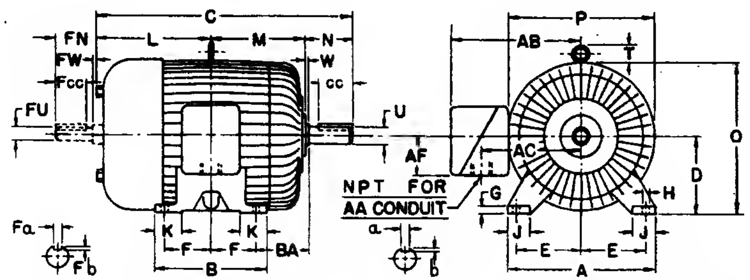
$$J = \frac{\pi}{2}(r_o^4 - r_i^4) = \frac{\pi}{2}(32 - 13.9) = \frac{\pi}{2^5}\, 18 = \cancel{}$$

$$\cancel{T = 8500} \times \frac{\pi}{\cancel{4}} \times \frac{18}{32} = 1.768$$

$$\tau = \frac{8{,}500 \times 1.1875}{1.768} = 5{,}700 \text{ psi}$$

4 HP. @ 1200 rpm.

PRINCIPAL DIMENSIONS OF Tamper



T.E.F.C. & T.E.F.C. EXPLOSION PROOF SQUIRREL-CAGE INDUCTION MOTORS
CEMA 182 TO 507US

THESE DIMENSIONS ARE FOR REFERENCE ONLY AND SHOULD NOT BE USED FOR CONSTRUCTION PURPOSES UNLESS CERTIFIED FOR A PARTICULAR ORDER (SEE OVER).
ALL DIMENSIONS IN INCHES.

TOLERANCES: "D" DIMENSION VARIES FROM +0 TO $-\frac{1}{32}$ FOR SIZES UP TO FRAME 326S INCLUSIVE, AND FROM +0 TO $-\frac{1}{16}$ FOR FRAMES ABOVE 326S.
"U" DIMENSION VARIES IN RANGE $\frac{3}{4}$ TO $1\frac{1}{2}$ INCL. +.0000 TO -.0005, ABOVE $1\frac{1}{2}$: +.000 TO -.001.

MOTORS MAY BE USED FOR FLOOR, WALL OR CEILING MOUNTING

APPROVED SINGLE SHAFT EXTENSION MOTOR

MOTOR WITH SINGLE SHAFT EXTENSION — GENERAL DIMENSIONS

FRAME	A	B	C	D	E	F	G	H	J	K	L	M	N-W	O	P	T	U	W	BA
182	8 3/4	5 3/4	13 3/4	4 1/2	3 3/4	2 1/4	1/2	13/32	1 3/8	1 5/8	6 1/2	4 7/8	2 1/4	9 1/4	9 1/2	—	7/8	1/8	2 3/4
184	8 3/4	6 3/4	14 3/4	4 1/2	3 3/4	2 3/4	1/2	13/32	1 3/8	1 5/8	7	5 3/8	2 1/4	9 1/4	9 1/2	—	7/8	1/8	2 3/4
213	10 1/4	7	17 1/8	5 1/4	4 1/4	2 3/4	1/2	13/32	1 3/4	2	7 7/8	6 1/8	3	10 5/8	10 3/4	—	1 1/8	1/8	3 1/2
215	10 1/4	8 1/2	18 5/8	5 1/4	4 1/4	3 1/2	1/2	13/32	1 3/4	2	8 5/8	6 7/8	3	10 5/8	10 3/4	—	1 1/8	1/8	3 1/2
254U	12 1/2	9 3/4	22 [illegible]	6 1/4	5	4 1/8	1/2	17/32	2	2 1/4	10 [illegible]	8 1/4	3 3/4	12 3/8	12 3/8	2 3/8	1 3/8	1/8	4 1/4
256U	12 1/2	11 1/2	24 1/4	6 1/4	5	5	1/2	17/32	2	2 1/4	11 1/4	9 1/8	3 3/4	12 3/8	12 3/8	2 3/8	1 3/8	1/8	4 1/4
284U	14	11	26 1/8	7	5 1/2	4 3/4	3/4	17/32	2 1/2	3	11 3/4	9 3/8	4 7/8	14	14	2 3/8	1 5/8	1/8	4 3/4
286U	14	12 1/2	27 5/8	7	5 1/2	5 1/2	3/4	17/32	2 1/2	3	12 1/2	10 1/8	4 7/8	14	14	2 3/8	1 5/8	1/8	4 3/4
324U	15 1/2	12 1/4	29	8	6 1/4	5 1/4	3/4	21/32	2 1/2	3	12 7/8	10 3/8	5 5/8	16	16	2 3/4	1 7/8	1/8	5 1/4
324S	15 1/2	12 1/4	26 5/8	8	6 1/4	5 1/4	3/4	21/32	2 1/2	3	12 7/8	10 3/8	3 1/4	16	16	2 3/4	1 5/8	1/8	5 1/4
326U	15 1/2	13 3/4	30 1/2	8	6 1/4	6	3/4	21/32	2 1/2	3	13 5/8	11 1/8	5 5/8	16	16	2 3/4	1 7/8	1/8	5 1/4
326S	15 1/2	13 3/4	28 1/8	8	6 1/4	6	3/4	21/32	2 1/2	3	13 5/8	11 1/8	3 1/4	16	16	2 3/4	1 5/8	1/8	5 1/4
364U	17 1/2	13 1/2	34 3/4	9	7	5 5/8	7/8	21/32	3	3 1/4	16 7/8	11 3/8	6 3/8	18 1/8	18 1/8	2	2 1/8	1/8	5 7/8
364US	17 1/2	13 1/2	32 1/8	9	7	5 5/8	7/8	21/32	3	3 1/4	16 7/8	11 3/8	3 3/4	18 1/8	18 1/8	2	1 7/8	1/8	5 7/8
365U	17 1/2	14 1/2	35 3/4	9	7	6 1/8	7/8	21/32	3	3 1/4	17 3/8	11 7/8	6 3/8	18 1/8	18 1/8	2	2 1/8	1/8	5 7/8
365US	17 1/2	14 1/2	33 1/8	9	7	6 1/8	7/8	21/32	3	3 1/4	17 3/8	11 7/8	3 3/4	18 1/8	18 1/8	2	1 7/8	1/8	5 7/8
404U	19 1/2	14 1/2	38 3/8	10	8	6 1/8	1	13/16	3	3 1/2	18 1/2	12 5/8	7 1/8	20 1/8	20 1/8	2 1/4	2 3/8	1/8	6 5/8
404US	19 1/2	14 1/2	35 1/2	10	8	6 1/8	1	13/16	3	3 1/2	18 1/2	12 5/8	4 1/4	20 1/8	20 1/8	2 1/4	2 1/8	1/8	6 5/8
405U	19 1/2	16	39 7/8	10	8	6 7/8	1	13/16	3	3 1/2	19 1/4	13 3/8	7 1/8	20 1/8	20 1/8	2 1/4	2 3/8	1/8	6 3/8
405US	19 1/2	16	37	10	8	6 7/8	1	13/16	3	3 1/2	19 1/4	13 3/8	4 1/4	20 1/8	20 1/8	2 1/4	2 1/8	1/8	6 5/8
444U	21 1/2	17 1/2	44	11	9	7 1/4	1 1/4	13/16	3 1/2	4	20 5/8	14 5/8	8 5/8	22 1/8	22 1/8	2 1/4	2 7/8	1/8	7 1/2
444US	21 1/2	17 1/2	39 5/8	11	9	7 1/4	1 1/4	13/16	3 1/2	4	20 5/8	14 5/8	4 1/4	22 1/8	22 1/8	2 1/4	2 1/8	1/8	7 1/2
445U	21 1/2	19 1/2	46	11	9	8 1/4	1 1/4	13/16	3 1/2	4	21 5/8	15 5/8	8 5/8	22 1/8	22 1/8	2 1/4	2 7/8	1/8	7 1/2
445US	21 1/2	19 1/2	41 5/8	11	9	8 1/4	1 1/4	13/16	3 1/2	4	21 5/8	15 5/8	4 1/4	22 1/8	22 1/8	2 1/4	2 1/8	1/8	7 1/2
505U	24	21 1/2	51 3/4	12 1/2	10	9	1 1/4	15/16	4	4 1/2	24 1/4	16 7/8	10	25 3/4	26 1/2	3	3 3/8	5/8	8 1/2
505US	24	21 1/2	46 3/4	12 1/2	10	9	1 1/4	15/16	4	4 1/2	24 1/4	16 7/8	5	25 3/4	26 1/2	3	2 5/8	5/8	8 1/2
507U	24	25 1/2	55 3/4	12 1/2	10	11	1 1/4	15/16	4	4 1/2	26 1/4	18 7/8	10	25 3/4	26 1/2	3	3 3/8	5/8	8 1/2
507US	24	25 1/2	50 3/4	12 1/2	10	11	1 1/4	15/16	4	4 1/2	26 1/4	18 7/8	5	25 3/4	26 1/2	3	2 5/8	5/8	8 1/2

KEYWAY, TERMINAL BOX, AND DATA FOR DOUBLE SHAFT EXT.

FRAME	Keyway Drive End a	Keyway Drive End b	Keyway Drive End cc	AA	Terminal Box Expl. Proof AB	Expl. Proof AC	Expl. Proof AF	Non-Expl. Pr. AB	Non-Expl. Pr. AC	Non-Expl. Pr. AF	Opp. End Shaft FN-FW	C+FN	FU	FW	Keyway Opp. End Fa	Fb	Fcc	FRAME
182	3/16	3/32	1 3/8	3/4	8 1/8	6	2 1/4	7 5/8	6 1/4	2 1/8	2 1/4	16 1/4	7/8	1/4	3/16	3/32	1 3/8	182
184	3/16	3/32	1 3/8	3/4	8 1/8	6	2 1/4	7 5/8	6 1/4	2 1/8	2 1/4	17 1/4	7/8	1/4	3/16	3/32	1 3/8	184
213	1/4	1/8	2	3/4	8 7/8	6 3/4	2 1/4	8 3/8	7	2 1/8	2 1/4	19 5/8	7/8	1/4	3/16	3/32	1 3/8	213
215	1/4	1/8	2	3/4	8 7/8	6 3/4	2 1/4	8 3/8	7	2 1/8	2 1/4	21 1/8	7/8	1/4	3/16	3/32	1 3/8	215
254U	5/16	5/32	2 3/4	1	11 3/4	8 3/4	3 1/8	10 1/2	8 3/8	2 3/4	3	25 5/8	1 1/8	1/8	1/4	1/8	2	254U
256U	5/16	5/32	2 3/4	1	11 3/4	8 3/4	3 1/8	10 1/2	8 3/8	2 3/4	3	27 3/8	1 1/8	1/8	1/4	1/8	2	256U
284U	3/8	3/16	3 3/4	1 1/4	12 1/2	9 1/2	3 1/8	11 7/8	9 1/2	2 7/8	3 3/4	30 1/8	1 3/8	1/4	5/16	5/32	2 3/4	284U
286U	3/8	3/16	3 3/4	1 1/4	12 1/2	9 1/2	3 1/8	11 7/8	9 1/2	2 7/8	3 3/4	31 5/8	1 3/8	1/4	5/16	5/32	2 3/4	286U
324U	1/2	1/4	4 1/4	1 1/2	13 7/8	10 5/8	3 3/8	13 1/4	10 5/8	3 3/8	4 7/8	34 1/8	1 5/8	1/4	3/8	3/16	3 3/4	324U
324S	3/8	3/16	1 7/8	1 1/2	13 7/8	10 5/8	3 3/8	13 1/4	10 5/8	3 3/8	3 1/4	30 1/8	1 5/8	1/4	3/8	3/16	1 7/8	324S
326U	1/2	1/4	4 1/4	1 1/2	13 7/8	10 5/8	3 3/8	13 1/4	10 5/8	3 3/8	4 7/8	35 5/8	1 5/8	1/4	3/8	3/16	3 3/4	326U
326S	3/8	3/16	1 7/8	1 1/2	13 7/8	10 5/8	3 3/8	13 1/4	10 5/8	3 3/8	3 1/4	31 5/8	1 5/8	1/4	5/8	3/16	1 7/8	326S
364U	1/2	1/4	5	2	15 1/2	11 7/8	3 5/8	14 1/2	11 7/8	3 5/8	5 3/8	40 1/2	1 7/8	1/8	1/2	1/4	4 1/4	364U
364US	1/2	1/4	2	2	15 1/2	11 7/8	3 5/8	14 1/2	11 7/8	3 5/8	3 3/4	36	1 7/8	1/8	1/2	1/4	2	364US
365U	1/2	1/4	5	2	15 1/2	11 7/8	3 5/8	14 1/2	11 7/8	3 5/8	5 5/8	41 1/2	1 7/8	1/8	1/2	1/4	4 1/4	365U
365US	1/2	1/4	2	2	15 1/2	11 7/8	3 5/8	14 1/2	11 7/8	3 5/8	3 3/4	37	1 7/8	1/8	1/2	1/4	2	365US
404U	5/8	5/16	5 1/2	2	17 1/4	12 3/8	4 1/2	15 7/8	12 3/8	4 1/2	6 3/8	44 7/8	2 1/8	1/8	1/2	1/4	5	404U
404US	1/2	1/4	2 3/4	2	17 1/4	12 3/8	4 1/2	15 7/8	12 3/8	4 1/2	4 1/4	39 7/8	2 1/8	1/8	1/2	1/4	2 3/4	404US
405U	5/8	5/16	5 1/2	2	17 1/4	12 3/8	4 1/2	15 7/8	12 3/8	4 1/2	6 3/8	46 3/8	2 1/8	1/8	1/2	1/4	5	405U
405US	1/2	1/4	2 3/4	2	17 1/4	12 3/8	4 1/2	15 7/8	12 3/8	4 1/2	4 1/4	41 3/8	2 1/8	1/8	1/2	1/4	2 3/4	405US
444U	3/4	3/8	7	2 1/2	18 1/4	13 3/4	4 1/2	16 7/8	13 3/4	4 1/2	7 1/8	51 1/4	2 3/8	1/8	5/8	5/16	5 1/2	444U
444US	1/2	1/4	2 3/4	2 1/2	18 1/4	13 3/4	4 1/2	16 7/8	13 3/4	4 1/2	4 1/4	44	2 1/8	1/8	1/2	1/4	2 3/4	444US
445U	3/4	3/8	7	2 1/2	18 1/4	13 3/4	4 1/2	16 7/8	13 3/4	4 1/2	7 1/8	53 1/4	2 3/8	1/8	5/8	5/16	5 1/2	445U
445US	1/2	1/4	2 3/4	2 1/2	18 1/4	13 3/4	4 1/2	16 7/8	13 3/4	4 1/2	4 1/4	46	2 1/8	1/8	1/2	1/4	2 3/4	445US
505U	7/8	7/16	8	3	20 3/4	16 7/8	5 5/8	21 1/8	17 1/8	5 5/8	10 5/8	62 3/8	2 5/8	5/8	5/8	5/16	8	505U
505US	5/8	3/16	3 1/2	3	20 3/4	16 7/8	5 5/8	21 1/8	17 1/8	5 5/8	5 5/8	52 3/8	2 3/8	5/8	5/8	5/16	3 1/2	505US
507U	7/8	7/16	8	3	20 3/4	16 7/8	5 5/8	21 1/8	17 1/8	5 5/8	10 5/8	66 3/8	2 5/8	5/8	5/8	5/16	8	507U
507US	5/8	5/16	3 1/2	3	20 3/4	16 7/8	5 5/8	21 1/8	17 1/8	5 5/8	5 5/8	56 3/8	2 3/8	5/8	5/8	5/16	3 1/2	507US

APPROVED DOUBLE SHAFT EXTENSION MOTOR

CERTIFIED FOR CUSTOMER

CUSTOMER ORDER No.

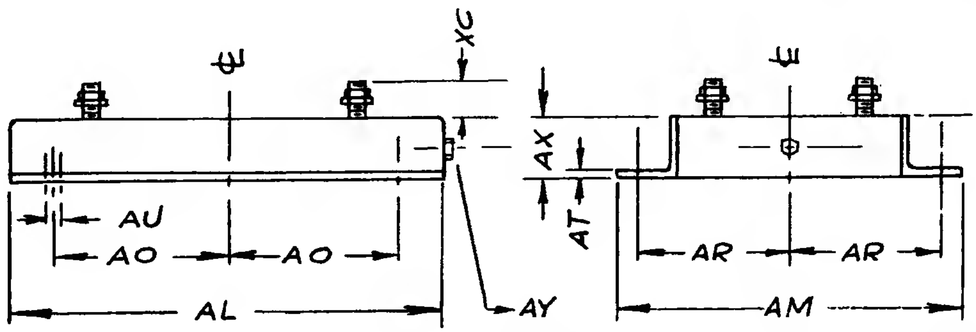
FRAME

APPROVED BY: P. ENG.

DATE

PRINCIPAL DIMENSIONS OF SLIDE BASES

Tamper DIVISION [redacted]
CANADA IRON FOUNDRIES LIMITED
160 ST. JOSEPH ST.
LACHINE P.Q. CANADA



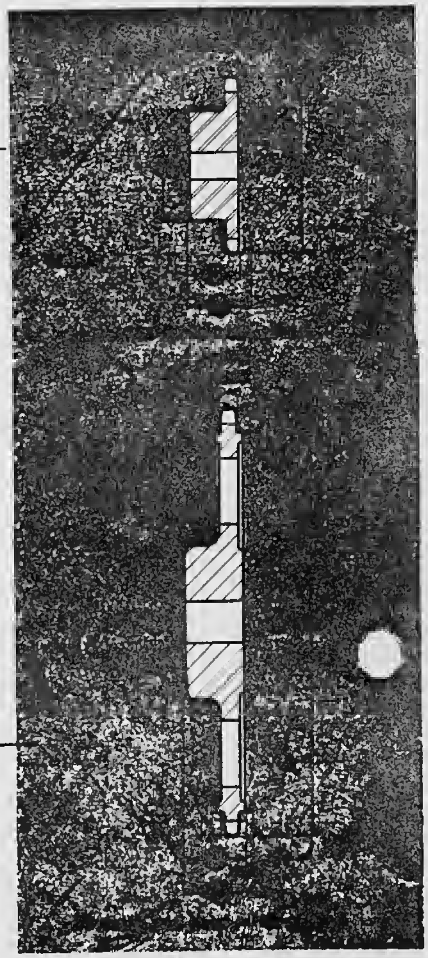
SIZE	AC	DC	AL	AM	AX	AO	AR	AU	BELT ADJUST.	AT	XC	D BOLT	AY BOLT	WEIGHT
66	*	*	12	8 1/2	1 1/8	4 1/2	3 7/8	5/8	3 1/2	.078	7/8	5/16 x 1	3/8 x 4	4
182	*		12 3/4	9 1/2	1 1/2	4 1/2	4 1/4	1/2	3	.134	1 1/4	3/8 x 1 1/2	1/2 x 5	9
184	*		12 3/4	10 1/2	1 1/2	4 1/2	4 3/4	1/2	3	.134	1 1/4	3/8 x 1 1/2	1/2 x 5	9 1/2
186		*	12 3/4	12	1 1/2	4 1/2	5 1/2	1/2	3	.134	1 1/4	3/8 x 1 1/2	1/2 x 5	11
187		*	12 3/4	13	1 1/2	4 1/2	6	1/2	3	.134	1 1/4	3/8 x 1 1/2	1/2 x 5	12
203	*	*	14	11	1 3/4	5	4 3/4	1/2	3	3/16	1 5/16	3/8 x 1 3/4	1/2 x 5	11 1/2
204	*	*	14	12	1 3/4	5	5 1/4	1/2	3	3/16	1 5/16	3/8 x 1 3/4	1/2 x 5	12
213	*		15	11	1 3/4	5 1/4	4 3/4	1/2	3 1/2	.164	1 1/4	3/8 x 1 1/2	1/2 x 5	13 1/2
215	*	*	15	12 1/2	1 3/4	5 1/4	5 1/2	1/2	3 1/2	.164	1 1/4	3/8 x 1 1/2	1/2 x 5	15 1/2
216		*	15	13 1/2	1 3/4	5 1/4	6	1/2	3 1/2	.164	1 1/4	3/8 x 1 1/2	1/2 x 5	17
224	*	*	15 1/2	12 1/4	1 3/4	5 1/2	5 3/8	1/2	3 1/2	3/16	1 5/16	3/8 x 1 3/4	1/2 x 5	13
225	*	*	15 1/2	13	1 3/4	5 1/2	5 3/4	1/2	3 1/2	3/16	1 5/16	3/8 x 1 3/4	1/2 x 5	14
254	*	*	17 3/4	15 7/8	2	6 1/4	6 5/8	5/8	4	3/16	1 5/16	1/2 x 1 3/4	5/8 x 6 1/2	17 1/2
256	*	*	17 3/4	16 7/8	2	6 1/4	7 1/2	5/8	4	3/16	1 5/16	1/2 x 1 3/4	5/8 x 6 1/2	18 1/2
284	*	*	19 3/4	16 7/8	2	7	7 1/2	5/8	4 1/2	3/16	1 5/16	1/2 x 1 3/4	5/8 x 6 1/2	21
286	*	*	19 3/4	18 3/4	2	7	8 1/4	5/8	4 1/2	3/16	1 5/16	1/2 x 1 3/4	5/8 x 6 1/2	22
324	*	*	22 3/4	19 1/4	2 1/2	8	8 1/2	3/4	5 1/4	3/16	2 3/16	5/8 x 2 1/2	3/4 x 9	31
326	*	*	22 3/4	20 3/4	2 1/2	8	9 1/4	3/4	5 1/4	3/16	2 3/16	5/8 x 2 1/2	3/4 x 9	32
364	*		25 1/2	20 1/2	2 1/2	9	9 1/8	3/4	6	1/4	2 1/8	5/8 x 2 1/2	3/4 x 9	44
365	*		25 1/2	21 1/2	2 1/2	9	9 5/8	3/4	6	1/4	2 1/8	5/8 x 2 1/2	3/4 x 9	45
404	*		28 3/4	22 3/8	3	10	9 7/8	7/8	7	1/4	2 3/8	3/4 x 2 3/4	3/4 x 9	60
405	*		28 3/4	23 7/8	3	10	10 5/8	7/8	7	1/4	2 3/8	3/4 x 2 3/4	3/4 x 9	61
444	*		31 1/4	24 5/8	3	11	11	7/8	7 1/2	5/16	2 5/16	3/4 x 2 3/4	3/4 x 11	83
445	*		31 1/4	26 5/8	3	11	12	7/8	7 1/2	5/16	2 5/16	3/4 x 2 3/4	3/4 x 11	85
504	*		35	28	3 1/2	12 1/2	12 1/2	1	8	5/16	3 1/16	7/8 x 3 1/2	7/8 x 16	150
505	*		35	30	3 1/2	12 1/2	13 1/2	1	8	5/16	3 1/16	7/8 x 3 1/2	7/8 x 16	152
583	*		38 3/4	29	4	14 1/2	13	1	8 1/2	3/8	3	7/8 x 3 1/2	7/8 x 16	178
584	*		38 3/4	31	4	14 1/2	14	1	8 1/2	3/8	3	7/8 x 3 1/2	7/8 x 16	182
585	*		38 3/4	33	4	14 1/2	15	1	8 1/2	3/8	3	7/8 x 3 1/2	7/8 x 16	185
586	*		38 3/4	35	4	14 1/2	16	1	8 1/2	3/8	3	7/8 x 3 1/2	7/8 x 16	189

ALL DIMENSIONS ARE IN INCHES, WEIGHT IN LBS. D-5.14 SEPT. 1959.
SUPERSEDES 2-21074/2 & 2-24788

A.S.A. 60 RENOLD Stock Sprockets - Single Strand

Type B

No. of teeth	Renold Catalogue No.	Pitch diameter	Top diameter (A)	Bore: Stock (B)	Bore: Maximum	Hub: Diameter (C)	Hub: Length through bore (D)	Chain track to hub face (E)
		in.	in.	in.	in	in.	in.	in.
9	211 753	2·924	3·58	1·00	1·250	2·27	1·55	2·12
10	211 754	3·236	3·84	1·00	1·500	2·58	1·55	2·12
11	211 755	3·549	4·11	1·00	1·750	2·89	1·75	2·32
12	211 756	3·864	4·39	1·00	2·00	3·21	1·75	2·32
13	211 757	4·179	4·78	1·00	2·00	3·02	1·75	2·32
14	211 758	4·494	5·08	1·00	2·125	3·35	1·75	2·32
15	211 759	4·810	5·37	1·00	2·375	3·67	1·75	2·32
16	211 760	5·126	5·67	1·00	2·625	3·99	1·75	2·32
17	211 761	5·442	5·98	1·00	2·75	4·125	1·75	2·32
18	211 762	5·759	6·28	1·00	2·75	4·125	1·75	2·32
19	211 763	6·076	6·59	1·25	3·00	4·50	2·00	2·57
20	211 764	6·392	7·00	1·25	3·00	4·50	2·00	2·57
21	211 765	6·709	7·31	1·25	3·00	4·50	2·00	2·57
22	211 766	7·027	7·62	1·25	3·00	4·50	2·00	2·57
23	211 767	7·344	7·93	1·25	3·00	4·50	2·00	2·57
24	211 768	7·661	8·24	1·25	3·00	4·50	2·00	2·57
25	211 769	7·979	8·56	1·25	3·00	4·50	2·00	2·57
26	211 770	8·296	8·87	1·25	3·00	4·50	2·00	2·57
30	211 774	9·567	10·13	1·50	2·75	5·00	2·25	2·58
32	211 775	10·202	10·76	1·50	2·75	5·00	2·25	2·58
35	211 777	11·156	11·70	1·50	2·75	5·00	2·25	2·58
36	211 778	11·474	12·02	[illegible]	[illegible]	5·00	2·25	2·58
40	211 780	12·746	13·28	1·50	3·00	5·50	3·00	3·33
45	211 783	14·336	14·86	1·50	3·00	5·50	3·00	3·33
48	211 785	15·290	15·81	1·50	3·00	5·50	3·50	3·83
54	211 788	17·198	17·72	1·50	3·00	5·50	3·50	3·83
60	211 792	19·107	19·62	1·75	3·25	6·00	3·50	3·83
70	211 797	22·289	22·79	1·75	3·25	6·00	3·75	4·08
80	211 802	25·471	25·97	1·75	3·50	6·50	3·75	4·08



In addition to the above stock range, sprockets in other sizes and for double, triple and quadruple strand chains, are available to order.

REBORING, KEYWAYING AND SETSCREWING
Stock Sprockets and Couplings

BORE TOLERANCES

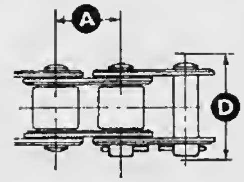
in.		in.
Up to 1 Dia.	=	Nom. to + ·001
1 to 2 Dia.	=	Nom. to + ·002
2 to 3 Dia.	=	Nom. to + ·003
3 to 4 Dia.	=	Nom. to + ·004
4 and up	=	Nom. to + ·005

KEYWAYS and SETSCREWS

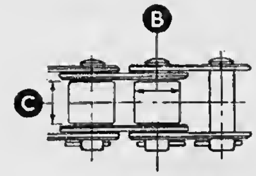
Shaft Diameter	Keyway *Width & Depth	Dia. of Set-screw	Shaft Diameter	Keyway *Width & Depth	Dia. of Set-screw	Shaft Diameter	Keyway *Width & Depth	Dia. of Set-screw
in.	in.	in.	in.	in.	in.	in.	in.	in.
$\frac{1}{2}-\frac{9}{16}$	$\frac{1}{8}\times\frac{1}{16}$	$\frac{1}{4}$	$1\frac{5}{16}-1\frac{3}{8}$	$\frac{5}{16}\times\frac{5}{32}$	$\frac{5}{16}$	$2\frac{13}{16}-3\frac{1}{4}$	$\frac{3}{4}\times\frac{3}{8}$	$\frac{3}{4}$
$\frac{5}{8}-\frac{7}{8}$	$\frac{3}{16}\times\frac{3}{32}$	$\frac{1}{4}$	$1\frac{7}{16}-1\frac{3}{4}$	$\frac{3}{8}\times\frac{3}{16}$	$\frac{3}{8}$	$3\frac{5}{16}-3\frac{3}{4}$	$\frac{7}{8}\times\frac{7}{16}$	$\frac{3}{4}$
$\frac{15}{16}-1\frac{1}{4}$	$\frac{1}{4}\times\frac{1}{8}$	$\frac{1}{4}$	$1\frac{13}{16}-2\frac{1}{4}$	$\frac{1}{2}\times\frac{1}{4}$	$\frac{1}{2}$	$3\frac{13}{16}-4\frac{1}{2}$	$1\times\frac{1}{2}$	$\frac{3}{4}$
			$2\frac{5}{16}-2\frac{3}{4}$	$\frac{5}{8}\times\frac{5}{16}$	$\frac{5}{8}$			

*Keyway width tolerances are plus ·002" minus ·000"; keyway depth tolerances are plus ·010" minus ·000". Standard taper keyways are tapered $\frac{1}{8}$" per foot. Taper keyway width tolerances are plus ·002" minus ·000"; depth tolerances are plus ·000" minus ·010".

RENOLD Single Strand Roller Chains A.S.A.



RIVETED



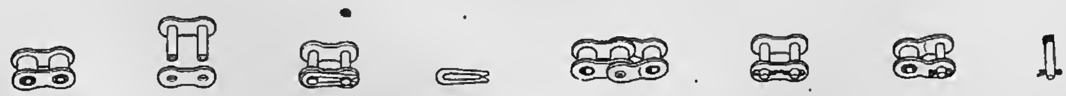
DETACHABLE

For further dimensions, see Designers Data Sheet – page 38

A.S.A. Chain No.	Pitch (A)	Roller diameter (B)	Bushing diameter	Between roller link plates (C)	Chain† track (D)	Nominal bearing area	Breaking load	Weight per foot	Renold Chain No.	SPARE PARTS AVAILABLE
RIVETED										
	in.	in.	in.	in.	in.	sq. in.	lb.	lb.		
25*	1/4		·130	1/8	·45	·017	800	·10	120 020	4, 107, 26, 27
35*	3/8		·20	3/16	·68	·041	2,250	·22	129 033	4, 107, 26, 27, 30, 59, 8
41	1/2	·306		1/4	·70	·049	2,000	·26	119 040	4, 107, 26, 27, 30, 59, 8
40	1/2	·312		5/16	·88	·068	4,000	·41	119 043	4, 107, 26, 27, 30, 59, 8
50	5/8	·40		3/8	1·14	·109	6,250	·66	119 053	4, 107, 26, 27, 30, 59, 8
60	3/4	·469		1/2	1·32	·163	8,500	1·01	119 063	4, 107, 26, 27, 30, 59, 8
80	1	·625		5/8	1·70	·275	14,500	1·68	119 083	4, 107, 58, 59, 8
100	1¼	·75		3/4	2·08	·40	24,000	2·47	119 103	4, 107, 58, 59, 8
120	1½	·875		1	2·58	·605	34,000	3·72	119 123	4, 107, 58, 59, 8
140	1¾	1·0		1	2·72	·725	46,000	5·05	119 143	4, 107, 58, 59, 8
160	2	1·125		1¼	3·24	·99	58,000	6·50	119 163	4, 107, 58, 59, 8
200	2½	1·562		1½	4·02	1·67	95,000	10·40	119 203	4, 107, 58, 59, 8
DETACHABLE										
60	3/4	·469		1/2	1·32	·163	8,500	1·07	119 064	4, 58, 59, 8
80	1	·625		5/8	1·70	·275	14,500	1·72	119 084	4, 58, 59, 8
100	1¼	·75		3/4	2·08	·40	24,000	2·54	119 104	4, 58, 59, 8
120	1½	·875		1	2·58	·605	34,000	3·91	119 124	4, 58, 59, 8
140	1¾	1·0		1	2·72	·725	46,000	5·12	119 144	4, 58, 59, 8
160	2	1·125		1¼	3·24	·99	58,000	6·64	119 164	4, 58, 59, 8
200	2½	1·562		1½	4·02	1·67	95,000	10·69	119 204	4, 58, 59, 8

* Rollerless chain
† See Data Sheet page 38 for definition

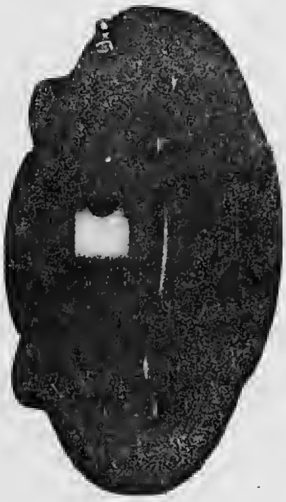
SPARE PARTS

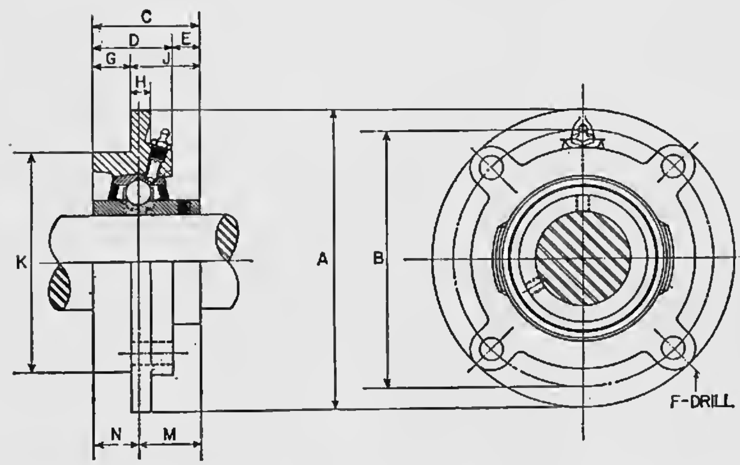


No. 4 Roller link — No. 107 Pin link — No. 26 Connecting link spring clip — No. 27 Spring clip fastener — No. 30 Two pitch offset link — No. 58 Connecting link double cotter — No. 59 Offset link — No. 8 Connecting pin and cotter (Only used with No. 59)

SEALMASTER BALL BEARINGS — MFC

MEDIUM DUTY FLANGE CARTRIDGE UNIT





[illegible]	②③ Unit No.	DIMENSIONS IN INCHES												Approx. Shipping Weight Lbs.
		A	B	C	D	E	F	G	H	J	K +.000 −.002	M	N	
15/16 1	MFC-15 MFC-16	4 3/8	3 6/8	1 9/16	1 3/16	3/8	3/8	1/4	3/8	1 6/16	3.000	7/8	11/16	3.0
1 3/16 1 1/4	MFC-19 MFC-20	5	4 1/8	1 11/16	1 1/4	7/16	7/16	3/8	3/8	1 5/16	3.375	1	11/16	3.9
1 7/16 1 1/2	MFC-23 ⑥MFC-24	5 1/4	4 3/8	1 15/16	1 15/32	15/32	7/16	7/16	7/16	1 1/2	3.625	1 3/16	3/4	4.7
1 1/2	MFC-24H	6 1/8	5 1/8	1 15/16	1 7/16	1/2	1/2	7/16	7/16	1 1/2	4.250	1 3/16	3/4	6.0
1 11/16 1 3/4	MFC-27 MFC-28	6 1/8	5 1/8	2 1/32	1 15/32	9/16	1/2	15/32	7/16	1 9/16	4.250	1 9/32	3/4	6.5
1 15/16 2	MFC-31 MFC-32	6 3/8	5 3/8	2 3/16	1 5/8	9/16	1/2	5/8	7/16	1 9/16	4.500	1 5/16	7/8	7.5
2 3/16 2 1/4	MFC-35 MFC-36	7 1/8	6	2 9/16	1 7/8	11/16	9/16	7/8	1/2	1 11/16	5.000	1 9/16	1	10.5
2 7/16 2 1/2	MFC-39 MFC-40	7 5/8	6 1/2	2 15/16	2 3/16	3/4	9/16	1 1/8	9/16	1 13/16	5.500	1 3/4	1 3/16	14.0
2 11/16	MFC-43	8 3/4	7 1/2	3 1/16	2 5/16	3/4	11/16	1 1/16	9/16	2	6.375	1 3/4	1 6/16	18.7
2 15/16 3	MFC-47 MFC-48	8 3/4	7 1/2	3 1/4	2 7/16	13/16	11/16	1 1/4	5/8	2	6.375	1 15/16	1 6/16	20.0
3 7/16 3 1/2	④MFC-55 ④MFC-56	10 1/4	8 6/8	3 25/32	2 3/4	1 1/32	13/16	1 1/8	3/4	2 21/32	7.375	2 7/32	1 9/16	32.0
3 15/16 4	④MFC-63 ④MFC-64	10 7/8	9 3/8	4 5/8	3 1/4	1 3/8	13/16	1 1/8	7/8	3 1/2	⑤8.125	2 11/16	1 15/16	44.0

① For load ratings, see page 13.
② For Contact Seals — use suffix "C".
③ Above units available in Millimeter Bores, see page 58.
④ Lubrication fitting on front face of casting.
⑤ Outside diameter tolerance +.000 —.005.
⑥ For load rating, see page 12.

ENGINEERING CASE LIBRARY

ECL 87C

Design of a Creme-Lite Mixer (C)

While Harry Anson was busy detailing the mixer and preparing the drawings, George addressed himself to Webb's inquiry about cooling the fat from 130°F to 80°F (Exhibit C-1).

For the analysis, George assumed that the cooling water temperature would be at 70°F, 10° below the required final temperature. As expected, the temperature was found to decrease exponentially. Some searching was required to determine suitable values for the specific heat of fat $S_H = .64$, and for the coefficients of heat transfer at the water interface h_o and the fat interface h_i.

The cooling time was found to be approximately 6 hours. Even with errors in the approximations for the heat transfer coefficients, this indicated that the fat could be cooled readily during the night shift. This information was passed to Webb at West Ltd.

Harry Anson completed the detail drawings of the tank (Exhibit C-2).

 Prepared for the Stanford Engineering Case Program by G. Kardos, Associate Professor of Mechanical Engineering, McMaster University, Hamilton, Ontario, Canada, with the support of the National Science Foundation. Although names of individuals and firms have been fictionalized, the case is based upon real engineering activity.

Before releasing them to the shop for manufacture, George checked them carefully to ensure that they met all the requirements. Harry had done an excellent job. Only a few minor additions were required. Of special note is the scraper assembly on DE-l: Harry had located standard hardware "Burndy Nyloclips" to attach the scrapers instead of having special attachments made as had been expected.

The tank was fabricated by the shop. During fabrication the shop, from its experience, added additional bracing to the legs. This was a point both Harry and George had missed. The addition added to the rigidity of the structure.

Before the tank was completed, Charlie and George went down to the shop to inspect it. They noticed that the welding of the pan containing the chain drive was discontinuous and on the inside. This was in line with accepted economical structure welding practice. But in this case it was unsatisfactory as food handling equipment. To prevent contaminants entering the fats, they asked the shop to put a continuous bead of weld on the outside of the joint.

The inspection also brought to the shop's attention areas where the welding had not been properly ground flush. These were re-worked. The shops hadn't any previous experience with food handling equipment and had a tendency to fall back on standard commercial practices.

George was now on his way to G. West Ltd. plant in response to Charlie's telephone call. Something had happened to the mixer. It had been installed only a couple of weeks earlier. George later related to Charlie what had happened.

"When I arrived at the plant, I was taken by Webb directly to their overhaul department. The gear box was apart. During the first trial of the mixer, the gear box had failed. Webb showed me the bearings and the worm from the box; they were badly scored and obviously had failed from an overload.

"Webb informed me that he had had the 'Crofts' representative in. They had concluded that the gear box had failed due to overloading. They had concluded that 5 Hp. motor with 3 to 4 belt drive would overload the gear box. Webb virtually accused me of having engineered it incorrectly.

"I was in a rather tight spot. I knew what was happening, but I had to explain it to Webb in a manner that he could comprehend and accept. He was obviously in a mood where he wanted to saddle us with all the repair charges.

"I told Webb that I agreed the gear box failed from overload. Then I pointed out that the 5 Hp. motor had been chosen because of its availability. I stated that even at the 20 rpm requirement, the mixer would only draw about 3 Hp., not enough to overload the gear box.

"I tried to find out whether the fat had been allowed to cool below 80°F or if mixing had been stopped for any reason. He assured me that it hadn't although from the vagueness of his reply I am certain it had. Of course, with Webb being the customer, I could not afford a direct confrontation. This was where it was less important to be right than to satisfy him.

"I assured him that at the actual operating speed, 15 rpm, the paddles just could not draw the necessary horsepower to overload the gear box. I inquired if a motor current reading had been taken during installation. Webb admitted they had taken readings with a 'clip-on' meter and that the current was negligible. I pointed out that this indicated that the motor was not running anywhere near full load.

"We were at an impasse as to fixing the blame. Therefore, we turned to the more important task of what to do to put the mixer back in service and prevent the failure from recurring.

"The gear box could be repaired by replacing the bearings. With a little pressing, Webb agreed that from their limited running experience the 15 rpm mixing speed

would be satisfactory. Webb also stated that their maintenance department had electric motors of various sizes available and the replacement of the 5 Hp. motor with a 3 Hp. motor could be readily done.

"It was then agreed that the unit would be put back in service with a 3 Hp. motor. I suggested that the motor be fused so that if more than 3 Hp. load was generated the fuse would blow.

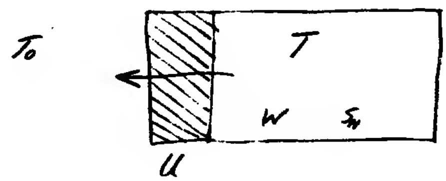
"Webb and I agreed that if operation of the mixer caused the fuses to blow, the design must be faulty and Roche Inc. would replace the gear box with a larger one. If the mixer functioned satisfactorily, the design must be satisfactory and no further action would be required on the part of Roche Inc."

George and Charlie informed the general manager of Roche Inc. about what happened at West Ltd. Nobody was happy with the situation, especially the prospect of having to replace the gear box.

Nothing further was heard from Webb. Charlie has had no occasion to call at G. West Ltd. since. The mixer is in service and operating satisfactorily. It can only be assumed that on the initial trials, the fat had been allowed to crystallize and in attempting to start the mixer up again, the drive system had been overloaded.

EXHIBIT C-1 GEORGE'S COOLING TIME CALCULATIONS

HEATING TIME



$T_a' = 70°F$ $T_o = 130$ @ $t = 0$ $T_f = 80$ @ $t = ?$

$$Q = \frac{dH}{dt} = U(T - T_a')$$

$$H = W \cdot S_H \, (\cancel{T - T_o}) \, (T_o - T)$$

$$-\frac{dH}{dt} = W S_H \frac{dT}{dt}$$

$$-W S_H \frac{dT}{dt} = U(T - T')$$

$$-\frac{dT}{(T-T')} = \frac{U}{W S_H} \, dt$$ let $\frac{U}{W S_H} = \alpha$

$$-\int \frac{dT}{(T-T')} = \alpha \int dt$$

$$-\Big[\ln \; T-T'\Big]_{T_o}^{T} = \alpha \Big[t\Big]_{t_o}^{t}$$

$$-\ln[T-T'] + \ln[T_o - T'] = -\alpha[t - t_o] + C$$

let $t = t_o \;\therefore\; T = T_o$

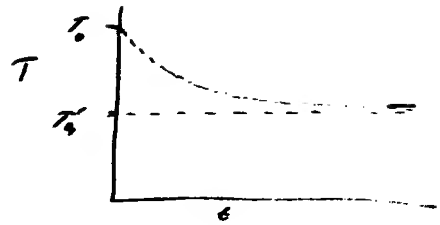
$$-\ln[T_o - T'] + \ln[T_o - T'] = \alpha[t_o - t_o] + c$$

$$\ln\left[\frac{T_o - T'}{T_o - T'}\right] = \cancel{\alpha} C$$

$$C = 0$$

EXHIBIT C-1 (Cont.)

$$\therefore \quad \frac{T_m - T'}{T_0 - T'} = e^{-\alpha t}$$



$$@\ t=0 \quad T - T' = T_0 - T'$$
$$T = T_0$$

$$@\ t=\infty \quad T - T' = 0$$

$$\text{Use} \quad \ln \frac{T - T'}{T_0 - T'} = -\alpha t$$

$$\ln \frac{T_0 - T'}{T_f - T'} = \alpha t \qquad \ln \frac{T_0 - T'}{T}$$

$$\alpha t_f = \ln(T_0 - T') - \ln[T_f - T']$$

$$T_0 - T' = 60^\circ F$$
$$T_f - T' = 10^\circ F$$

$$\ln 60 = 4.09434$$
$$\ln 10 = 2.30259$$
$$\alpha t_f = 1.79175$$

$$\alpha = \frac{U}{W S_H} \qquad W = 5400\ \#, \quad S_H = .64$$

$$U = u \times A$$

$$A = \pi \times 5 \times 5 + \frac{\pi}{4} \times 5^2 = \frac{5\pi}{4} \times 25 = 98.4 \text{ ft Sq.}$$

say 100 sq ft.

$$u = \frac{1}{\frac{1}{h_o} + \frac{x}{K} + \frac{1}{h_i}}$$

$$h_o = 300, \quad h_i = 10, \quad x = .1875, \quad K = 105$$

EXHIBIT C-1 (Cont.)

$$\frac{1}{300} + \frac{.1875}{105} + \frac{1}{10}$$

$$\frac{1}{f} = .003 + .0018 + .1 = .1048$$

$$u = 10$$

$$\alpha = \frac{10 \times \cancel{1000}\ 100}{5400 \times .64} = \frac{10}{34.6} = .29$$

$$1.79175 = .29\ t$$

$$t = \frac{1.79}{.29} = 6.17$$

Say 6 hrs.

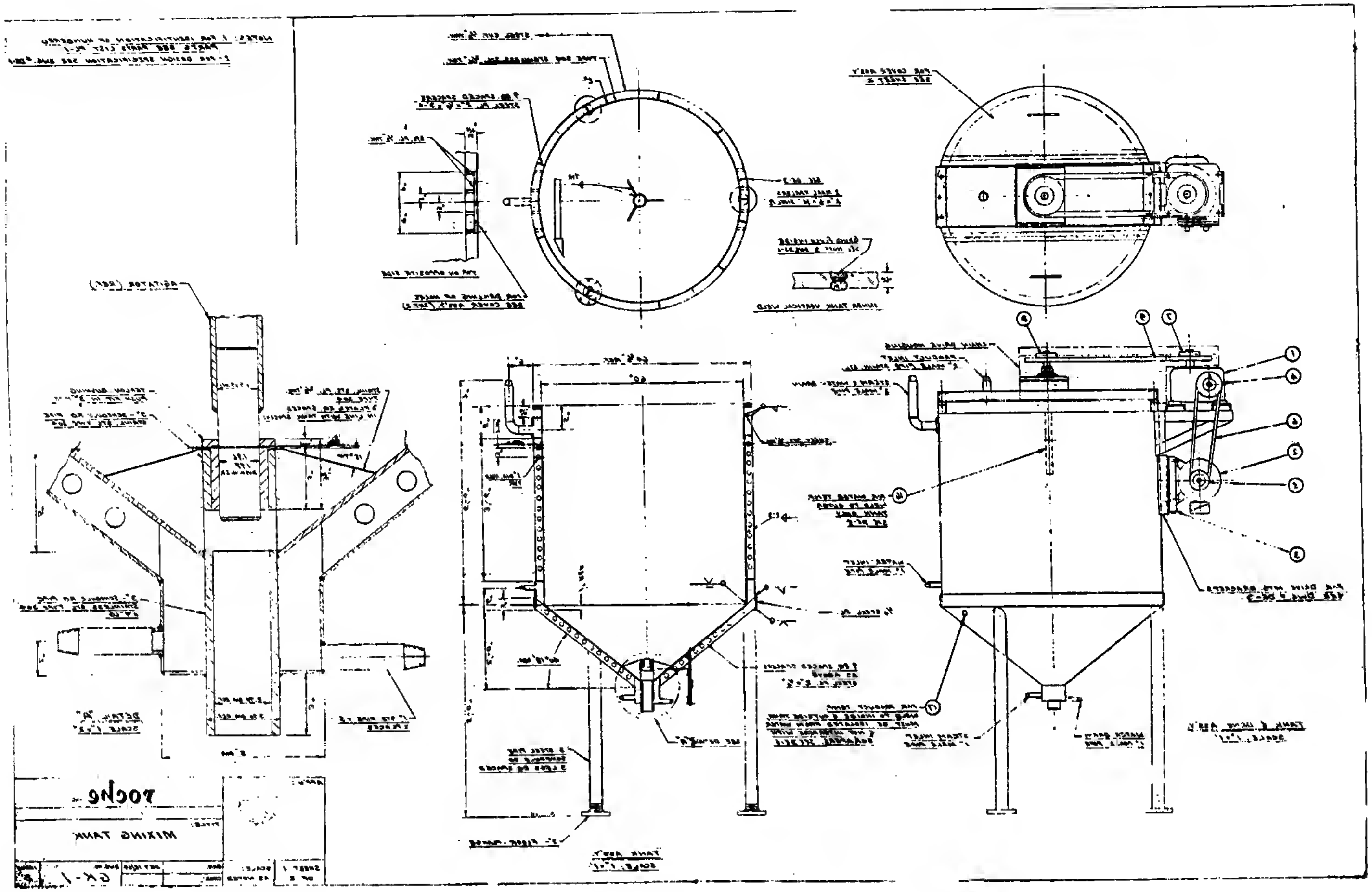


EXHIBIT C-2 DETAIL DRAWING

Sheet 1 of 5

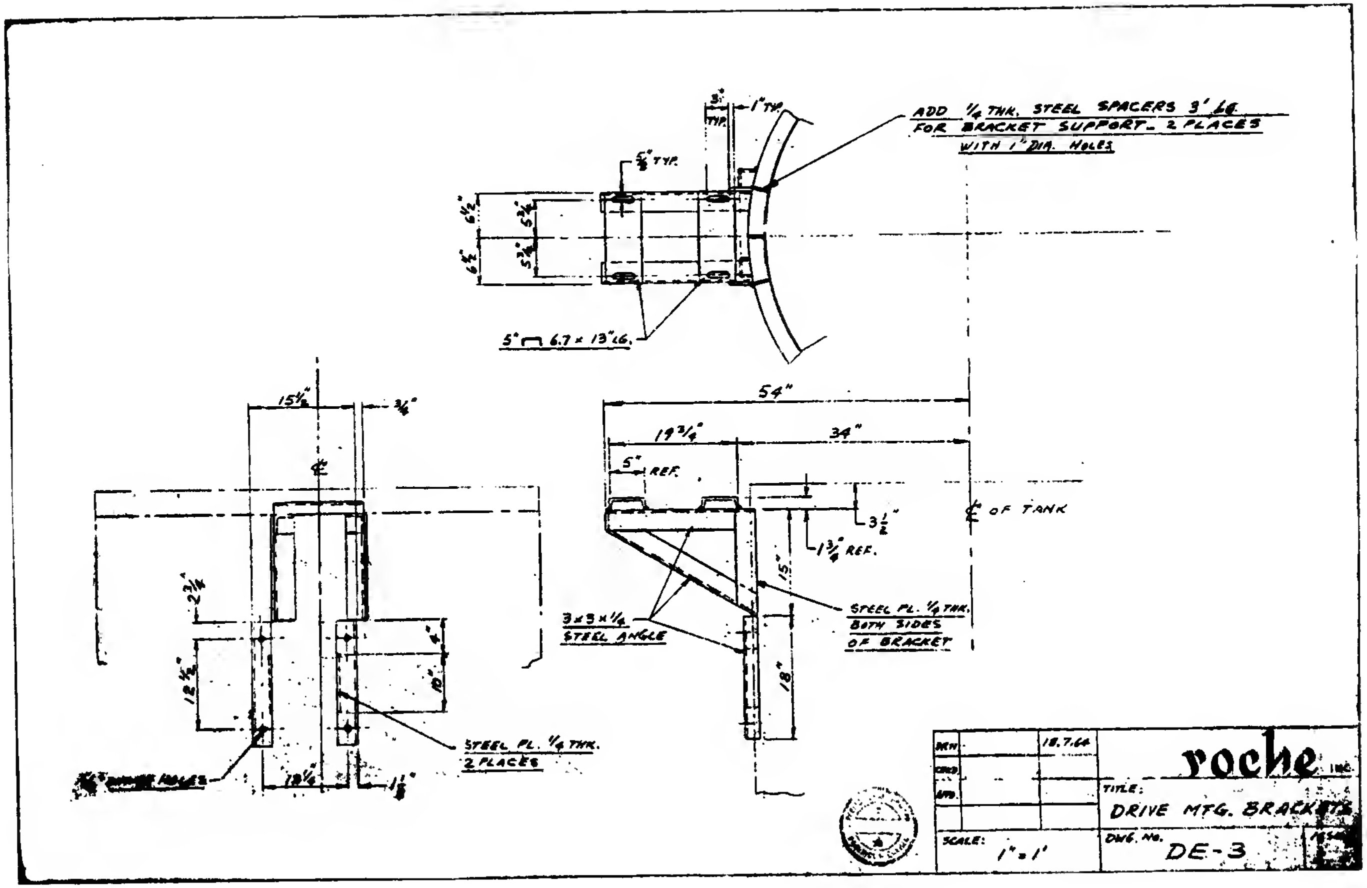


EXHIBIT C-2 (Cont.)

Sheet 2 of 5

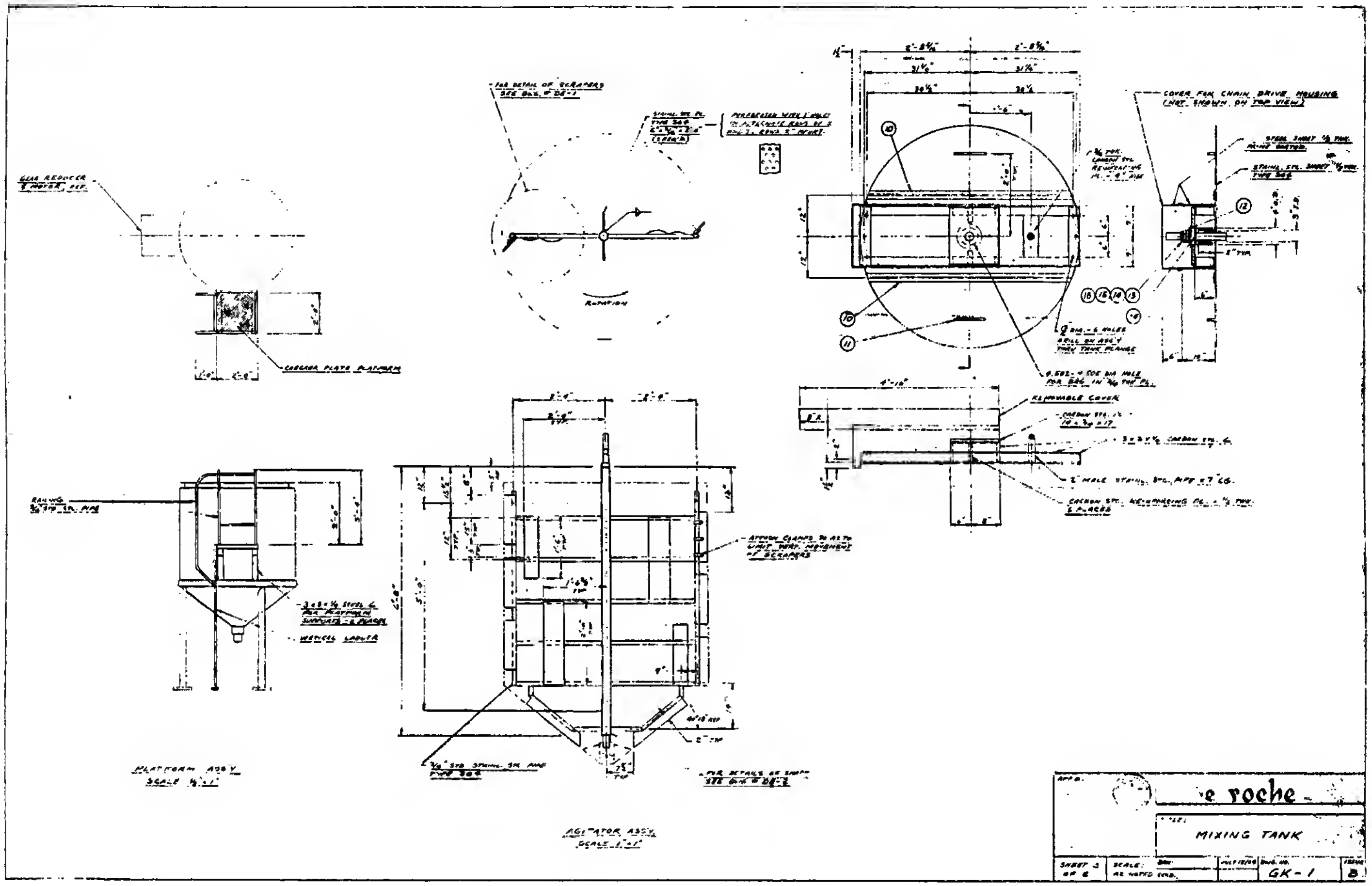


EXHIBIT C-2 (Cont.)

Sheet 3 of 5

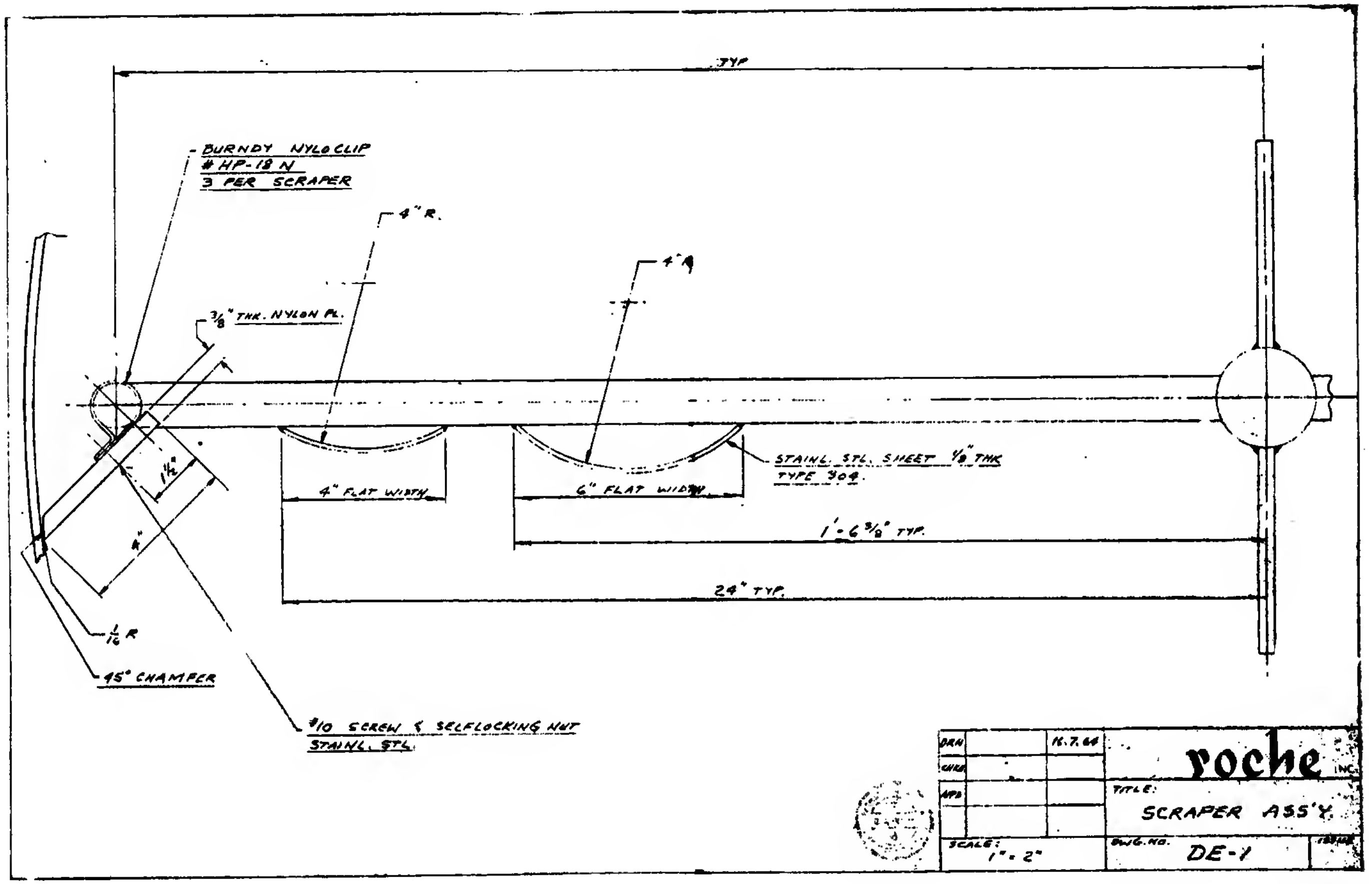


EXHIBIT C-2 (Cont.)

Sheet 4 of 5

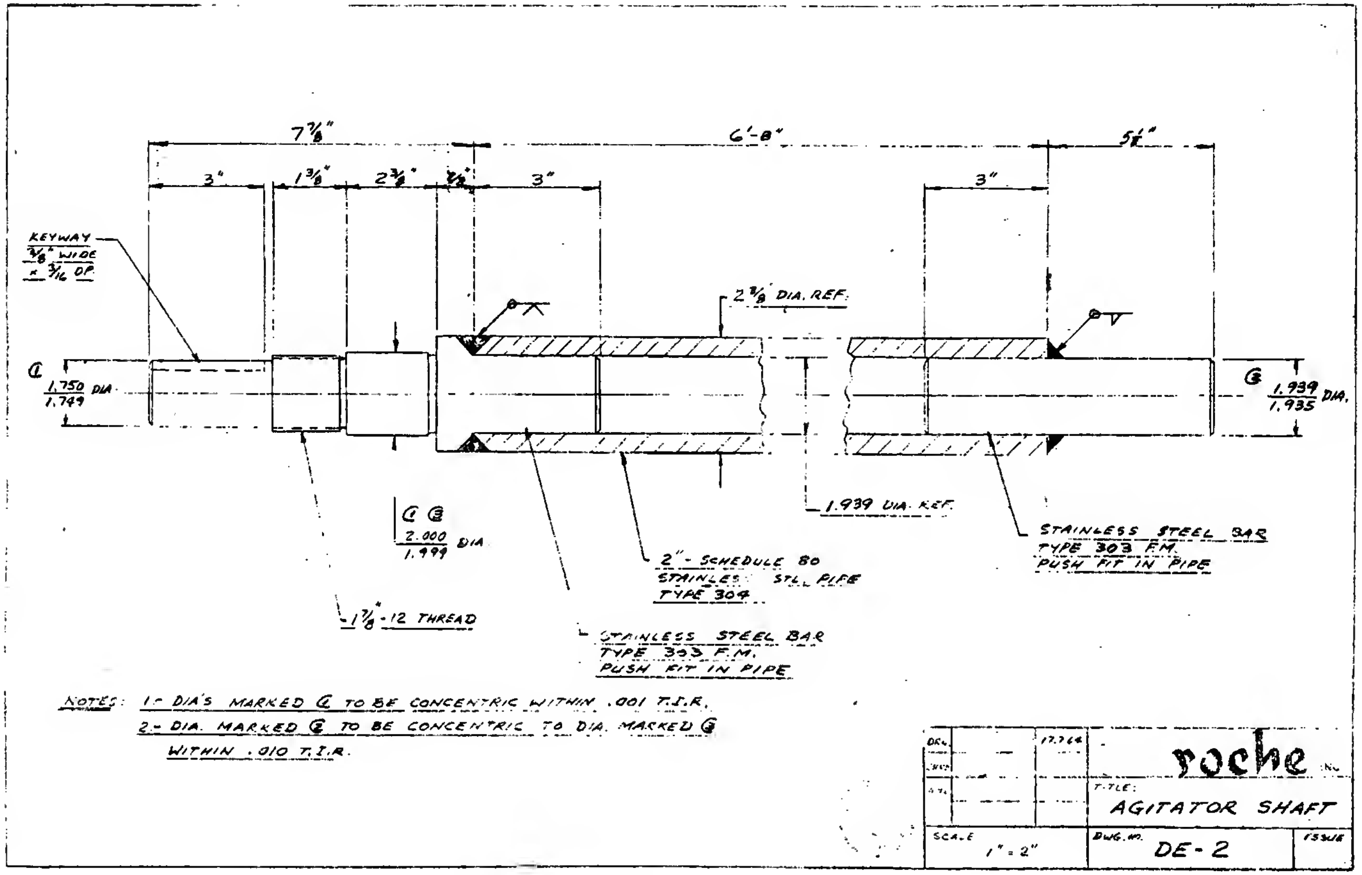


EXHIBIT C-2

Sheet 5 of 5

Design of a Creme-Lite Mixer

This case is intended to show the student the full spectrum of design engineering activity from requirement definition to trouble-shooting in the field. Of special interest is the fact that the case demonstrates the use of several engineering sciences in an elementary way. The division of the case into three parts permits the assignment of projects, the results of which can be later compared with the results in the case.

The exhibits in the case demonstrate how the practicing engineer uses the various engineering subjects in making technical decisions.

Exhibit B-I	Pressure Vessel Design
Exhibit B-I	Fluid Dynamic Drag
Exhibit B-3	Elementary Calculus
Exhibit B-5	Torsional Stressing
Exhibit B-5	Motor, Chain, Bearing Selection
Exhibit C-I	Heat Transfer

Part A

This segment of the case deals with problem definition. Class discussion can be centered around:

- Food handling equipment special requirements
- Vagueness of performance requirements
- Why viscosity of creme-lite is required
- George's method of getting the "feel" of the creme-lite
- Determination of power requirements
- Mixer paddle sizing and arrangement
- Tank design for 10 psi

In addition to the above, the student can be assigned the task of determining the power requirements of the mixer and sketching a paddle arrangement. It should be impressed upon the student that the paddle arrangement selection should be made on a rational basis, not just appearance.

Part B

This segment, with its exhibits, demonstrates the solution to the requirements that George arrived at. Close scrutiny of the exhibits will reveal his rationale for the decisions that resulted in the final design. Besides comparing the assignment against George's solution, class discussion can be centered around:

The relative merits of the two drag calculations made; one using average velocity, the other integrating over paddle faces.

Significance of Reynolds number.

The reasons for selecting paddle sizes to produce zero force on the central shaft.

Tank wall thickness was determined both from elementary strength of materials and ASME code; what are the merits of each and how was the final design established.

The motor selection.

The selection of V belt and chain drives.

How to calculate the time it will take to cool a tank of creme-lite from 130°F to 80°F.

Parts A and B may be used as an assignment in graphics. Based upon the design details given, the student could be asked to make layouts or finished drawings of the mixer tank.

Part C

This segment of the case contains the finished drawing of the tank and the mixer and the heat transfer calculations. The body of the case outlines the subsequent field service problems. The class discussions could be centered around:

- Merits of the design.
- What caused the gear box failure.
- George's handling of the problem.
- Should the unit have been originally supplied with 3 hp. motors.

Assignments could be made in graphics to make drawings of some missing mixer details.